# PRODUCT ENGINEERING

VOLUME 7 NUMBER 8

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## September Number

The feature article "Comparative Tests of a Hydraulically-Driven and a Gear-Driven Lathe" by Max Kronenberg, Dr. Eng., is a comprehensive investigation. It deals with the performance, under braking and cutting tests, of these two types of drives, with observations covering chip-volume, energy input and output, and variations in efficiency and capacity caused by changes of speed.

An article by W. M. Griffith covering the many changes and additions that have been made in materials and design standards, since the publication of his "Engineering Standards for the Design of Springs" in the July 1933 number of *P. E.* This supplementary data will be welcomed by all designers interested in spring applications.

In line with the attention given by designers of domestic products to the problem of noise suppression in household appliances, S. C. Eaker in "Silencing the Modern Refrigerator" discusses the development of a design to avoid the generation of noise.

An article for machine designers, "The Relation Between Size, Configuration and Characteristics" by Cyril Donaldson, treats of the general problem of designing a new machine to replace an old one that has been satisfactory. It also outlines a method for determining the relationship existing between the characteristics of two machines having similar geometrical proportions.

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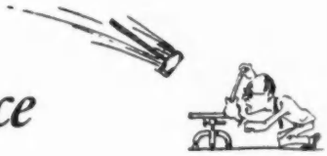
# *Intimate Correspondence*



## Where to Get Drafting Standards

PRODUCT ENGINEERING
NEW YORK, N. Y.

With reference to your July number, page 261, of the article "Drafting Room Methods and Management," by J. K. Olsen, may I please ask where I can obtain a copy of the book "Drawing and Drafting Room Practice,'" mentioned in this article. It also states that the book is issued by the American Standards Association. —W.D.C.

*Marysville, Calif.*

THIS book can be obtained from the American Standards Association, 29 West 39th Street, New York City. The price of the book is 50 cents and goes under the No. 314.1-1935. It might also be mentioned that a similar book "Drawing and Drafting Room Practice—Graphical Symbols," is also available. The price of this book is 45 cents and its number designation is 314.2-1935.

## Bourdon Tubes

PRODUCT ENGINEERING
NEW YORK, N. Y.

I have been a reader of your magazine for some years, but do not recall ever having seen information published on the subject of bourdon tubes. The type of article that I have in mind is one which would give formulas and curves to show the relationship between the wall thickness, width, depth of the tube section, the included angle of the helix, and deflection for any working pressure. Also, there is need for information on determining the elastic limit of a tube in terms of internal pressure. In other words, an article that would tell one how to proceed in determining these various variables for any required deflection per unit pressure. I think such an article would be helpful to many designers. —P.L.S.

*Milwaukee, Wis.*

THE information requested will be found scattered among various text books. Perhaps some of our readers have some ideas on this subject, and we will be glad to hear from designing engineers who have made a special study of this subject.

## Advertisers, Please Note

PRODUCT ENGINEERING
NEW YORK, N. Y.

I have noticed that some advertisers in *Product Engineering*, and other publications as well, call attention to only one of their products in an advertisement. Others may direct the reader's attention to one or two products in each advertisement, but use a small portion of the ad to let the reader know what else they make. I often find this additional information convenient and I wonder why the advertisers rarely give it. —C.J.

*Toledo, Ohio*

FRANKLY, we editors are not expert on advertising copy writing, nor are we authorities on the motives that dictate the type of copy used and the scope of the advertisement. But, it would be interesting to get further reader reactions on the thought expressed by C. J. Unquestionably, the advertising pages of a magazine are a source of information just as well as the editorial pages.

## Just a Boner

PRODUCT ENGINEERING
NEW YORK, N. Y.

In the June number of *Product Engineering*, second line from the top of the middle column you state that the manufacturer announces a "complimentary" line of controls and valves for use with the new pumps. I am just wondering whether he is giving these away as premiums with every pump sold, or whether the word "complementary" should have been used? I suspect the latter. But then why should a publication that preaches technical accuracy make such a literal mistake. —W.E.C.

*Philadelphia, Pa.*

OUR correspondent is entirely correct in berating us for such an untimely error. The part that we cannot understand is that we have not heard from the Hydraulic Press Manufacturing Company, accusing us of giving out misleading information. And our only excuse is that mistakes will happen.

## What is Durhy?

PRODUCT ENGINEERING
NEW YORK, N. Y.

Recently I saw an electrical resistant material mentioned as being "new to the trade, known as Durhy, one of the carborundums which possesses . . . . also the additional properties in that it offers decreasing electrical resistance as its temperature rises."

I would be very much interested to find out who makes this material and if you have any information about it I would appreciate very much hearing from you. —B.E.H.

*Springfield, Mass.*

FRANKLY, we have hunted high and low for Durhy, but have been unable to find out who makes it or details about its composition. Of course, the easiest thing to do is to ask our readers. We would like to help out B.F.H. and would appreciate any clues or information about Durhy.

## Page Mr. Olsen, Please

PRODUCT ENGINEERING
NEW YORK, N. Y.

I am deeply interested in the discussion "Drafting Room Methods and Management," by Mr. J. K. Olsen, formerly chief draftsman of the Stewart-Warner Corporation, appearing in the July issue of *Product Engineering*. I would like to communicate Mr. Olsen and inasmuch as I am not in possession of his mailing address would appreciate it if you would send it to me. —T.H.S.

*Muskegon, Mich.*

WE are always glad to furnish addresses of authors of articles appearing in *Product Engineering*. For the benefit of others who might like to have Mr. Olsen's address, it is 2313 W. Diversey Ave., Chicago, Ill. We might add that we have had quite a number of other letters requesting information relative to his articles.

## Comments on the Index

FROM various chief engineers and executives we have received numerous comments concerning the "Where-to-Buy" directory that appears in June *Product Engineering*. Such comments included:

"Directory is of real value" . . . . "I think it is a very useful compilation" . . . . "I am of the opinion that this is a commendable addition to your magazine and should prove helpful to designers and engineers" . . . . "Accept my compliments on a good magazine" . . . . "Certain of the items are very important in our business, and we will make good use of it" . . . . "I find the directory especially helpful in locating novelty items and unusual applications."

Of course, all comments are not complimentary to the extent of those quoted above. Some contain ideas to further improve the contents of our next directory. The writer of the following letter points out a criticism and makes a suggestion to which we are giving immediate attention. He writes:

"Such a directory should certainly be of considerable value to many firms and individuals when it has reached the point of covering more fields and types of equipment. I hope that some future issue will contain a more complete listing and classification. We feel it would be highly desirable to arrange to furnish a bound volume of such a completed directory as it would be very valuable and time saving. Of course you are familiar with the fact that there are other directories along these same general lines, but they are not classified in a manner most convenient and informative for the designing engineer."

# PRODUCT ENGINEERING

GEORGE F. NORDENHOLT, *Editor*

MCGRAW-HILL PUBLISHING CO., INC. AUGUST 1936 VOLUME SEVEN, NUMBER EIGHT


## *More Engineering or More Material?*

IN THE dim past, when reinforced concrete was in its infancy, there was a designer of concrete structures who looked with doubt upon the combination of steel and concrete. He admitted one day that he played safe by putting enough concrete into his structures so that they would be well within safe limits whether the steel took any of the load or not.

Somewhat the same attitude has been adopted by oil well equipment builders until very recently. They are making tools for a group that is nearer the pioneer stage than any other branch of industry. Grit and stamina under hard conditions have been more characteristic of the oil field workers than have engineering and technical skill. Their equipment was designed originally to work in smaller and much shallower holes than are common today, and in consequence it is now subjected to far more strenuous conditions.

Until the engineer began to get into the picture the answer to a structural failure was to add more material to the section that broke down. Then, when that recourse led to tools so ponderous they could hardly be manipulated, the manufacturers went to better and costlier alloy steels. Today hardly any other material is to be found in an oil rig.

The next step, which is now being taken, is the redesign of sections to insure that all the material in them is used to the best advantage. This sort of engineering requires technical ability rather than reliance on rule of thumb, and represents a departure from the methods of the pioneer.

Older industries have reached this stage in their development long since, and with it a better understanding of what the competent engineer has to offer. In the machine tool industry, for example, careful tests of frame and unit deflections under load have shown how far from efficient performance some of the most successful models really were.

In welded construction something of the same development process took place. Early welded structural joints were patterned after riveted joints and fell short of the efficiencies they should have exhibited. Machine bases welded from steel shapes followed the lines of iron castings and were needlessly complicated and expensive until the engineer was called in to analyze and improve them.

WHAT is generally true of an industry is not necessarily true of all the plants in that industry. Some companies have used engineering talent to avoid the wastes of traditional designs. Others have persisted in the old ways, too stubborn or too stupid to take advantage of opportunities.

It is hardly necessary to tell product engineers that extra material is no effective substitute for designing brains, but they should lose no chance to impress this fact on the managements of their companies. Acceptance of the principle means more opportunities for engineers and engineering, but at the same time it leads to better products, bigger sales and fatter profits.

# *Deflections and Stresses In COILED SPRINGS Made of Rectangular Wire*

E. W. STEWART
*Vice-President, The William D. Gibson Company*

**Discussion of standard and basic formulas from which are developed convenient design equations and charts with the constants checked and modified by actual test data**

IN SEARCHING for a basis on which rectangular wire springs could be figured, it was found that the great French mathematician, St. Venant, had published in 1855 an analysis of stress and deflection in a straight bar of rectangular section. This analysis is given in Todhunter and Pearson's "A History of the Elasticity and Strength of Materials" Vol. II, Part I, page 39, published in 1893 by Cambridge University Press.

A good analysis of the subject was also found in a paper entitled, "Safety Valve Springs" by William L. DeBeaufre published in the May 1917 number of the *Journal of the American Society of Naval Engineers.* This paper was the first published application of St. Venant's theory to the calculation of springs, as far as it has been determined. Later the same derivations from St. Venant's original work were given in a slightly different form by Timoshenko in his book "Strength of Materials," Vol. I, page 77, published by D. Van Nostrand.

In any compression or extension spring, the principal stresses are torsional. Hence in a spring of rectangular stock it makes no difference in the calculations whether the material is coiled "on edge," with the longer side at right angles to the axis of the helix, or on "flat side," with the longer side parallel to the axis. This can readily be visualized if one considers a rectangular bar or shaft being twisted. Its resistance to twist would be the same no matter at what angle to the vertical its longer axis might be placed.

For load and deflection of a rectangular bar in torsion, the basic formula is as follows:

$$f = \left(\frac{I_1 + I_2}{4I_1 I_2}\right)\left(\frac{MRL}{KG}\right) \qquad (1)$$

in which

$f$ = deflection in in. along circumference of radius $R$
$I_1$ = rectangular moment of inertia about one axis of the section
$I_2$ = rectangular moment of inertia about the axis at right angles to $I_1$
$M$ = torsional moment, load times radius of application = $WR$
$R$ = radius at which load is applied
$L$ = length of bar in in.
$G$ = torsional modulus of rigidity
$K$ = a constant which varies with the shape of section

In a coiled spring $R = D/2$ = mean radius of coils.

***Fig. 1a—Cross-section of wire before coiling and Fig. 1b after coiling***

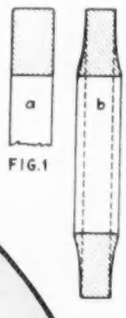
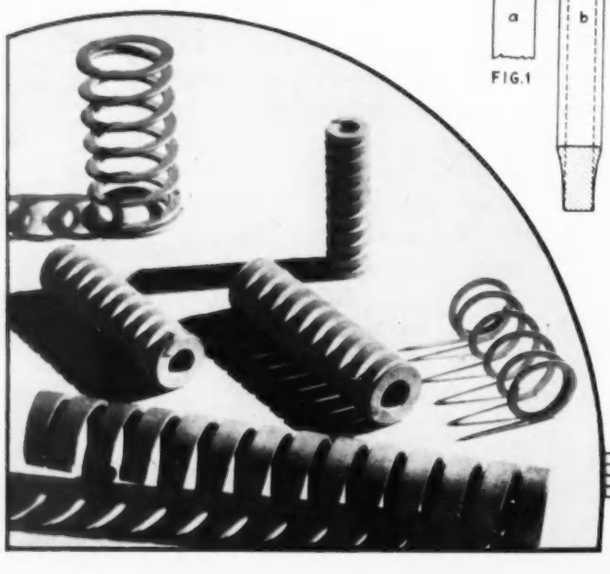



FIG. 1



$L = 2\pi NR$ where $N$ is the number of active coils, $b$ = width or maximum dimension of the rectangular section, and $t$ = its thickness in inches. These may be substituted directly in Equation (1), which gives the following equation:

$$f = \left[\frac{(bt^3/12) + (b^3t/12)}{4(bt^3/12)\ (b^3t/12)}\right]\left[\frac{(WD/2)\ (D/2)\ (2\pi DN/2)}{KG}\right] \quad (2)$$

$$f = \left[3\left(\frac{b^2 + t^2}{b^3t^3}\right)\right]\left[\frac{2\pi}{8K}\left(\frac{WD^3N}{G}\right)\right] \quad (3)$$

$$f = \frac{6\pi}{8K}\left(\frac{(b^2 + t^2)\ WD^3N}{Gb^3t^3}\right) \quad (4)$$

In these equations $K$ varies from a value of 0.843462 when $b/t = 1$, to a value of unity when the ratio of $b/t$ is infinity. St. Venant's tabulation gives the following values of $K$ for the corresponding ratios of $b/t$ as follows:

| $b/t$ = | 1 | 2 | 3 | 4 |
|---|---|---|---|---|
| $K$ = | 0.8435 | 0.8576 | 0.8777 | 0.8951 |
| $b/t$ = | 6 | 8 | 10 | 20 |
| $K$ = | 0.9198 | 0.9356 | 0.9463 | 0.9709 |

To eliminate the summation of squares and simplify Equation (4), since the load must vary as the product of the first power of the width times the cube of thickness, these being the factors in the moment of inertia of a rectangle, the formula may be rearranged thus:

$$f = \frac{3\pi}{4K}\left(\frac{b^2+t^2}{b^2}\right)\frac{WND^3}{Gbt^3} \quad (5)$$

$$= \frac{3\pi}{4K}\left(\frac{(b/t)^2 + 1}{(b/t)^2}\right)\frac{WND^3}{Gbt^3} \quad (6)$$

Then for any ratio of $b/t = R$, the constant for any ratio may be stated as

$$\frac{3\pi}{4K}\ \frac{(R^2+1)}{R^2} \quad (7)$$

When working with a slide rule, it is preferred to invert this expression, as it results in a constant less than unity in the numerator of the formula. Hence the constant which removes the sum of the squares, as based on St. Venant's analysis and which may be called $K_2$, or the reciprocal of Equation (7), becomes in more usable form

$$K_2 = \frac{4K}{3\pi}\left(\frac{R^2}{R^2 + 1}\right) \quad (8)$$

Equation (8) gives values of $K_2$ as follows:

| $b/t = R$ = | 1 | 2 | 3 | 4 |
|---|---|---|---|---|
| $K_2$ = | 0.1795 | 0.2912 | 0.3354 | 0.3570 |
| $b/t = R$ = | 6 | 8 | 10 | 20 |
| $K_2$ = | 0.3798 | 0.3910 | 0.3976 | 0.4110 |

Using the constant $K_2$, Equation (6) becomes:

$$f = \frac{1}{K_2}\left(\frac{WND^3}{Gbt^3}\right) \quad (9)$$

Using these constants in Equation (9) gives calculated loads approximating those obtained by tests. They would probably be in closer agreement in springs in which the section after coiling was truly rectangular, that is, with the longer sides parallel. They do check to practice quite well when $b/t$ is 8 or more, as the thickening of the material at the inside of the coil causes a less departure from a condition of parallel surfaces than when the ratio $b/t$ is less.

By far the greater percentage of springs of this type

**TABLE II—Conversion Factor for Round to Rectangular Wire Springs**

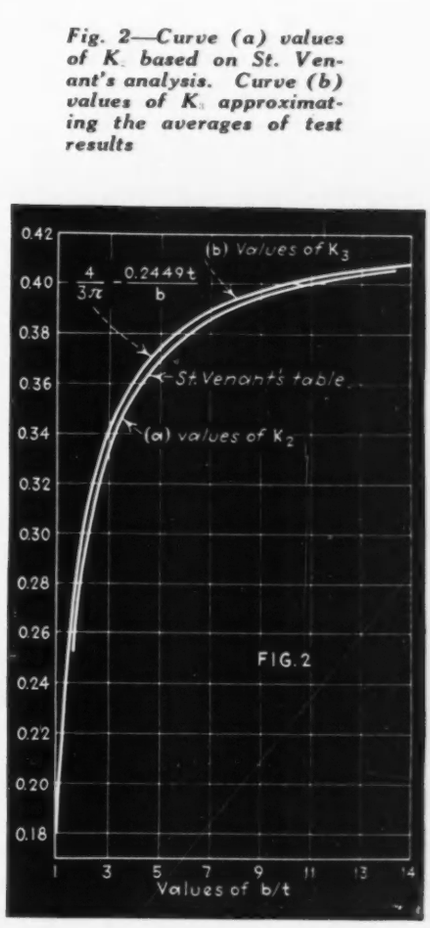
| $R$ | $X$ | $R$ | $X$ | $R$ | $X$ |
|---|---|---|---|---|---|
| 1.0 | 1.436 | 2.6 | 6.864 | 7.0 | 21.806 |
| 1.1 | 1.771 | 2.7 | 7.204 | 7.5 | 23.508 |
| 1.2 | 2.111 | 2.8 | 7.542 | 8.0 | 25.197 |
| 1.3 | 2.450 | 2.9 | 7.883 | 9.0 | 28.598 |
| 1.4 | 2.790 | 3.0 | 8.222 | 10 | 31.984 |
| 1.5 | 3.130 | 3.1 | 8.561 | 11 | 35.384 |
| 1.6 | 3.469 | 3.2 | 8.901 | 12 | 38.794 |
| 1.7 | 3.808 | 3.3 | 9.240 | 13 | 42.172 |
| 1.8 | 4.166 | 3.4 | 9.580 | 14 | 45.667 |
| 1.9 | 4.488 | 3.5 | 9.920 | 15 | 48.960 |
| 2.0 | 4.827 | 4.0 | 11.616 | 16 | 52.365 |
| 2.1 | 5.168 | 4.5 | 13.313 | 17 | 55.760 |
| 2.2 | 5.491 | 5.0 | 15.012 | 18 | 59.155 |
| 2.3 | 5.846 | 5.5 | 16.711 | 19 | 62.548 |
| 2.4 | 6.186 | 6.0 | 18.408 | 20 | 65.953 |
| 2.5 | 6.524 | 6.5 | 20.103 | | |

**TABLE I**

**Values of $K_3$ and $b/t$**

| $b/t$ | $K_3$ | $b/t$ | $K_3$ |
|---|---|---|---|
| 1.0 | 0.1795 | 3.4 | 0.3522 |
| 1.1 | 0.2013 | 3.5 | 0.3543 |
| 1.2 | 0.2199 | 4.0 | 0.3630 |
| 1.3 | 0.2356 | 4.5 | 0.3698 |
| 1.4 | 0.2491 | 5.0 | 0.3753 |
| 1.5 | 0.2608 | 5.5 | 0.3798 |
| 1.6 | 0.2710 | 6.0 | 0.3835 |
| 1.7 | 0.2800 | 6.5 | 0.3866 |
| 1.8 | 0.2881 | 7.0 | 0.3894 |
| 1.9 | 0.2953 | 7.5 | 0.3918 |
| 2.0 | 0.3017 | 8 | 0.3937 |
| 2.1 | 0.3076 | 9 | 0.3972 |
| 2.2 | 0.3120 | 10 | 0.3998 |
| 2.3 | 0.3177 | 11 | 0.4021 |
| 2.4 | 0.3222 | 12 | 0.4041 |
| 2.5 | 0.3262 | 13 | 0.4055 |
| 2.6 | 0.3300 | 14 | 0.4068 |
| 2.7 | 0.3335 | 15 | 0.4080 |
| 2.8 | 0.3367 | 16 | 0.4091 |
| 2.9 | 0.3398 | 17 | 0.4100 |
| 3.0 | 0.3426 | 18 | 0.4108 |
| 3.1 | 0.3452 | 19 | 0.4115 |
| 3.2 | 0.3477 | 20 | 0.4122 |
| 3.3 | 0.3500 | | |

***Fig. 2—Curve (a) values of $K_2$ based on St. Venant's analysis. Curve (b) values of $K_3$ approximating the averages of test results***



are of such proportions that $b/t$ varies from about 1.5 to about 5, and the ratio of $D/b$ from about 4 to about 8, and are made from wire which is rolled with the sides parallel before coiling. The wire upsets or increases in thickness at the inside of the coil and draws to a lesser thickness at the outside. The actual cross-section becomes distorted from that shown in Fig. 1*a* before coiling to more nearly that of Fig. 1*b*, which shows the section after coiling. If this distortion were such that the center of gravity of the section remained at the mean diameter of the coil the constants above would be correct as the mean thickness of the trapezoidal section would equal the uniform thickness of the wire before coiling.

The true center of gravity of the section after coiling is somewhat closer to the coil axis than the mean radius, hence the true neutral axis in torsion is at a somewhat shorter moment arm than the mean radius. In a spring of which all finished dimensions can be accurately measured, the theoretical constants can be correctly applied by locating the diameter upon which the true neutral axis lies by means of the following formula:

$$D_1 = ID + \frac{2b}{3}\left(\frac{2t_o + t_i}{t_o + t_i}\right) \qquad (10)$$

in which

$D_1$ = coil diameter at center of gravity of wire section
$ID$ = inside diameter of coil
$t_o$ = thickness of material at outside, as coiled
$t_i$ = thickness of material at inside, as coiled

In a proposed spring in the process of being designed, however, it is difficult to locate the diameter $D_1$, although it can be approximated by determining $t_i$ with reasonable accuracy for ordinary design problems by the formula:

$$t_i = 0.48t\left(\frac{ID + 2t}{ID + t} + 1\right) \qquad (11)$$

With $t_i$ calculated, it is then assumed that $t_o$ is as much less than the original uniform thickness as $t_i$ is greater, which is only approximately true. It is a rather involved matter when it is considered that there is no direct way to determine both $b$ and $t$ for a proposed spring design, and that several combinations may have to be set up by "cut-and-try" methods before arriving at final dimensions.

Over a long period test data has been compiled for reference of springs of this type actually made, and as previously stated the proportions are usually in the range where $D/b$ is 4 to 8 and $b/t$ is 1.5 to 5. Fig. 2 shows two curves, the lower being the values of $K_2$ plotted to the constants already given as derived from the theory and the upper curve showing values which may be called $K_3$, plotted to approximate the averages of a large number of test results. It has been found that one can use the constants corresponding to the upper curve without attempting to figure the true mean radius to the center of gravity of the coiled section, with

**TABLE III—Values of $K_4$ for Various Values of $b/t$**

| $b/t$ | $K_4$ | $b/t$ | $K_4$ |
|---|---|---|---|
| 1.0 | 0.416 | 3.6 | 0.561 |
| 1.1 | 0.431 | 3.7 | 0.563 |
| 1.2 | 0.442 | 3.8 | 0.565 |
| 1.3 | 0.452 | 3.9 | 0.568 |
| 1.4 | 0.460 | 4.0 | 0.571 |
| 1.5 | 0.467 | 4.5 | 0.581 |
| 1.6 | 0.474 | 5.0 | 0.590 |
| 1.7 | 0.480 | 5.5 | 0.597 |
| 1.8 | 0.486 | 6.0 | 0.603 |
| 1.9 | 0.492 | 6.5 | 0.609 |
| 2.0 | 0.498 | 7.0 | 0.614 |
| 2.1 | 0.503 | 7.5 | 0.611 |
| 2.2 | 0.508 | 8.0 | 0.620 |
| 2.3 | 0.513 | 9 | 0.626 |
| 2.4 | 0.518 | 10 | 0.631 |
| 2.5 | 0.523 | 11 | 0.635 |
| 2.6 | 0.527 | 12 | 0.638 |
| 2.7 | 0.531 | 13 | 0.640 |
| 2.8 | 0.535 | 14 | 0.642 |
| 2.9 | 0.539 | 15 | 0.644 |
| 3.0 | 0.542 | 16 | 0.645 |
| 3.1 | 0.546 | 17 | 0.646 |
| 3.2 | 0.549 | 18 | 0.647 |
| 3.3 | 0.552 | 19 | 0.648 |
| 3.4 | 0.555 | 20 | 0.649 |
| 3.5 | 0.558 | | |

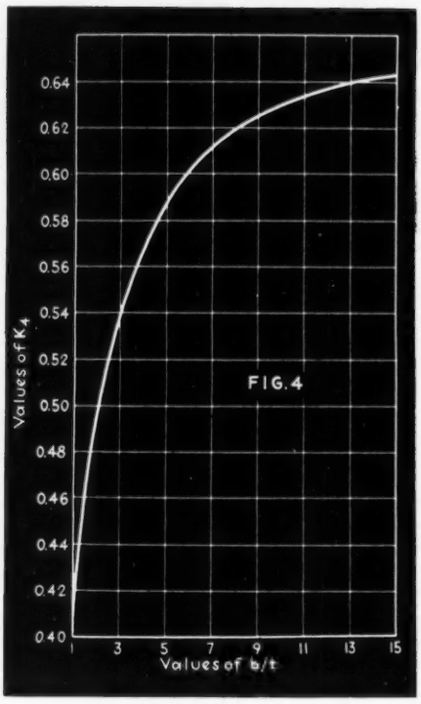



*Fig. 4—Graph of values of $K_4$ as given in Table III*

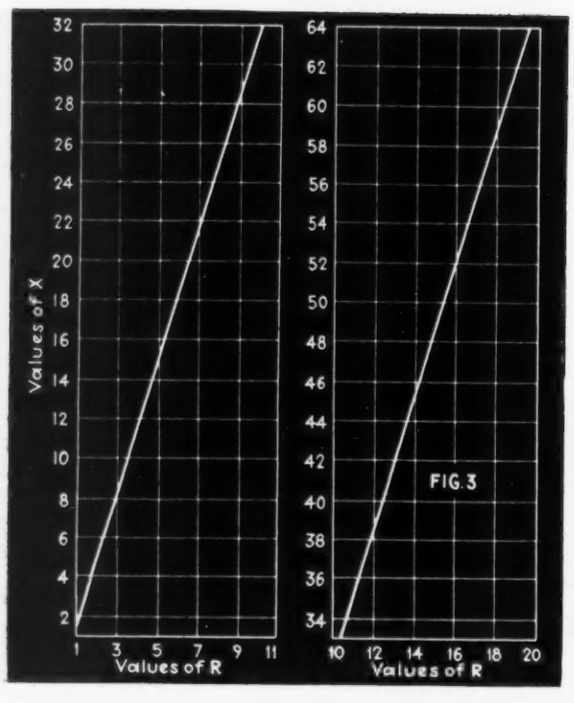



*Fig. 3—Graph of values of X as given in Table II*

agreement with tests within satisfactory commercial limits.

It will be noted that the two curves merge in the larger ratios of $b/t$, where the center of gravity approaches closely to the center of the width of the section, hence practically coincide with the mean diameter of the coil. The values of the upper curve are expressed by the equation.

$$K_3 = \frac{4}{3\pi} - \frac{0.2449t}{b} \qquad (12)$$

In the accompanying Table I the values of $K_3$ have been extended for all ratios of $b/t$ within probable practical use.

Obviously charts for springs of rectangular material would be complicated because of the fact that every ratio of $b/t$ has a different constant. To avoid this difficulty Table II has been worked out which gives the factors for converting from round-wire to rectangular-wire springs.

As an example, assume a round-wire spring in which the mean diameter and the number of active coils are the same as in a rectangular-wire spring under consideration and in which the size of round wire is the same as the thickness, or shorter side, of the rectangular section. The load-rate is then determined, or $W/f$ value, for the assumed round wire spring. This value multiplied by the conversion factor corresponding to the $b/t$ ratio in the rectangular wire, is the $W/f$ value for the spring of rectangular wire. This factor is established thus: if $X$ is the desired conversion factor, in a spring of rectangular material:

$$W = \frac{K_3 f G b t^3}{N D^3} \qquad (13)$$

In the assumed spring of round wire:

$$W = \frac{fGt^4}{8ND^3} \qquad (14)$$

Then

$$\frac{K_3 f G b t^3}{N D^3} = X\left(\frac{fGt^4}{8ND^3}\right) \qquad (15)$$

Substituting $R$ for the ratio $b/t$, R can be substituted for $b$, and $t$ becomes unity. Therefore, eliminating like terms

$$K_3 R = \frac{X}{8}, \text{ and } X = 8\ K_3 R \qquad (16)$$

In Fig 3 is a graph of the value of $X$, which are also given in Table II.

As an example of the use of Table II, to find the rate per in. of a compression spring 2 in. mean diameter and having 10 active coils of $\frac{1}{2}$ in. $\times$ $\frac{1}{8}$ in. rectangular steel wire. By reference to a chart or other wise, it is determined that a spring of $\frac{1}{8}$ in. round wire, 2 in. mean diameter and 10 active coils has a rate per in. deflection of 4.4 lb. The $b/t$ ratio is 4 for $\frac{1}{2}$ in. $\times$ $\frac{1}{8}$ in., for which the conversion factor is 11.616, as given in Table II. Multiplying 11.616 $\times$ 4.4 we get 51.11 lb. per in. deflection for the spring of rectangular wire.

St. Venant also established the theory upon which calculation of the torsional stress in springs of this type may be based. His analysis shows that when a rectangular bar is twisted the corners are stressed less than any other part of the section, the maximum stress is at the surface at the center of the long side, and an approximate expression for this maximum stress is:

$$S = \frac{WD}{2bt^2}\left(3 + \frac{1.8t}{b}\right) \qquad (17)$$

A table has been calculated for the constants $K_4$ for determining the maximum stress for various values of $b/t$. The values as given in Table III, are not based upon the preceding approximation, but upon St. Venant's tabulation of actual values, for the formula

$$S = \frac{WD}{K_4 bt^2} \qquad (18)$$

The various values of $K_4$ are also shown in the curve in Fig. 4.

The stress indicated by Equation (18) is torsional only, and does not take into account any correction for increased shear caused by curvature, as developed by Dr. A. M. Wahl for springs of round wire and pub-

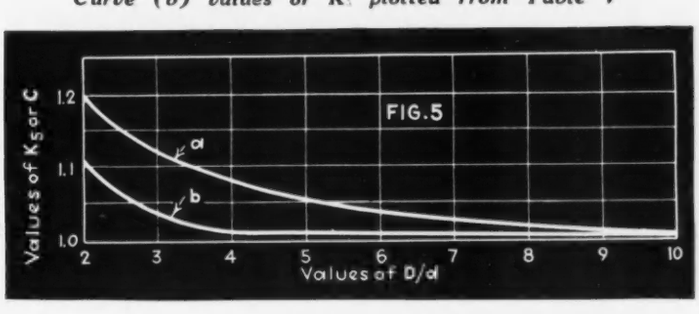
**TABLE IV—Correction Factor $C$ for $t_1 = cd$**

| $D/d$ | $C$ | $D/d$ | $C$ | $D/d$ | $C$ |
|---|---|---|---|---|---|
| 2.0 | 1.200 | 3.1 | 1.114 | 5.0 | 1.056 |
| 2.1 | 1.187 | 3.2 | 1.109 | 5.5 | 1.047 |
| 2.2 | 1.176 | 3.3 | 1.105 | 6.0 | 1.040 |
| 2.3 | 1.168 | 3.4 | 1.101 | 6.5 | 1.034 |
| 2.4 | 1.160 | 3.5 | 1.097 | 7.0 | 1.029 |
| 2.5 | 1.153 | 3.6 | 1.093 | 7.5 | 1.024 |
| 2.6 | 1.145 | 3.7 | 1.090 | 8.0 | 1.020 |
| 2.7 | 1.138 | 3.8 | 1.087 | 8.5 | 1.016 |
| 2.8 | 1.131 | 3.9 | 1.084 | 9.0 | 1.012 |
| 2.9 | 1.125 | 4.0 | 1.080 | 9.5 | 1.010 |
| 3.0 | 1.119 | 4.5 | 1.068 | 10.0 | 1.008 |

**TABLE V—Correction Factor for Increased Load per Unit Deflection in Springs Wherein $D/d$ is Small**

| $D/d$ | $K_5$ | $D/d$ | $K_5$ | $D/d$ | $K_5$ |
|---|---|---|---|---|---|
| 2.0 | 1.105 | 2.7 | 1.055 | 3.4 | 1.026 |
| 2.1 | 1.094 | 2.8 | 1.049 | 3.5 | 1.023 |
| 2.2 | 1.086 | 2.9 | 1.044 | 4.0 | 1.014 |
| 2.3 | 1.080 | 3.0 | 1.040 | 4.5 | 1.010 |
| 2.4 | 1.074 | 3.1 | 1.036 | 5.0 | 1.008 |
| 2.5 | 1.067 | 3.2 | 1.032 | 5.5 | 1.007 |
| 2.6 | 1.061 | 3.3 | 1.029 | 6.0 | 1.006 |

***Fig. 5—Curve (a) values of C as given by Table IV Curve (b) values of $K_5$ plotted from Table V***



lished in the 1928 *Transactions* of the American Society of Mechanical Engineers. This correction factor can be applied to rectangular wire springs which are coiled with the longer side of the wire parallel with the axis of the coil if the thickness $t$ is used to establish the spring index as $D/t$, corresponding to $C$ in Dr. Wahl's formula for the stress correction factor, which is:

$$S_1 = S\left(\frac{4C-1}{4C-4}+\frac{0.615}{C}\right) \tag{19}$$

There has been no analysis published, as far as the author knows, of what such a correction factor might or should be for rectangular wire coiled on edge, and there is no solution for that problem. Experience has developed the fact, however, that the maximum stress at maximum operating loads on such springs should never exceed 90,000 lb. per sq. in. as figured without such a correction factor and that increased life is obtained when the maximum stress is less than this figure.

The calculations of stress by these constants have been checked in the following manner. Round carbon steel wire of various sizes was coiled into springs in which the pitch was such that the elastic limit would be exceeded before the coils could close together. The springs were then hardened in oil and carefully drawn to a hardness of 44 Rockwell. Stress-strain test graphs were then plotted to determine the yield point in terms of axial loads on the springs. The stress for these elastic limit loads was then figured in terms of simple torsional stress from the well-known formula,

$$S = \frac{8DW}{\pi d^3} \tag{20}$$

Pieces of the same material from the same bundles of wire were then rolled to several ratios of $b/t$ and coiled into springs, both "on edge" and "on flat side." They were given the same heat treatment exactly, tested and the test results plotted to determine the elastic limit loads. The yield-point stresses corresponding to these loads as figured by the constants in Table III were in close agreement with the elastic limit stresses in the round-wire springs.

Because limitation of space is usually the prime reason for using square-wire springs they are too often made with the ratio of coil diameter to wire size smaller than is good practice, with no allowance made for the large increase in stress at the inside of the coil because of this small ratio. Also the upset increases the load on the spring for a given deflection when the ratio of coil diameter to wire size is small, thus further increasing the maximum stress over that calculated by the ordinary formulas, because the true neutral axis moves toward the center of the coil. For square wire of thickness $d$, the effective outside diameter $D_1$ is

$$D_1 = 1D + \frac{2d}{3}\left(\frac{2t_0+t_i}{t_0+t_i}\right) \tag{21}$$

For square wire, assuming that $t_o$ is approximately as much less than $d$ as $t_i$ is greater, and substituting $R$ for the spring index, or $D/d$, then:

$$D_1 = d\left[(R+0.33)-0.16\left(2+\frac{1}{R}\right)\right] \tag{22}$$

To obtain the solid height of such springs, which is the total number of coils multiplied by $t_i$, or slightly less depending upon how much the end coils are ground, the accompanying Table IV and curve $(a)$ shown in Fig. 5 provides a correction factor by which to multiply the original size $d$ to obtain $t_i$ the inside thickness.

For small ratios of $D/d$, the change in center of gravity of the section has been reduced to terms of increase in load per unit deflection. A correction factor called $K_3$ is shown in curve $(b)$, Fig. 5, and Table V. Using this correction factor, the load can be expressed thus:

$$W = \frac{0.1795\ K_3 fGd^4}{ND^3} \tag{23}$$

From this the stress for any load can be figured as:

$$S = \frac{WD}{0.416bt^2}\left(\frac{4C-1}{4C-4}+\frac{0.615}{C}\right) \tag{24}$$

in which $C = D/d$

The maximum pitch which will permit closing the spring without permanent set, depends upon the material. For steel the maximum stress should be 100,000 lb. per sq. in. and should never exceed 130,000 lb. per sq. in. when calculated by equations that include the correction factor. The life of such a spring will depend upon, the maximum stress at the greatest deflection as operated, and the range of stress between the maximum and the stress imposed by any initial load which may be stored in the spring at the starting end of its operating cycle.

In Fig. 6 is given a fatigue chart, based upon a large number of test runs, indicating what may be expected of oil tempered carbon steel wire S.A.E. X1065.

As an example of the use of Fig. 6, assume a spring made of S.A.E. 1065 steel to be stressed to 50,000 lb. per sq.in. by a preload, to find the stress up to which it can then be safely loaded at end of cycle read up on the 50,000 lb. per sq.in. ordinate (horizontal scale) to the endurance limit stress line, values of which are given on vertical scale, and read 86,000 lb. per sq.in. as stress at the finishing end of operation.

Space limitations sometimes make it impossible to design the spring to work within limits of allowable stress. In such instances fatigue breakdown is inevitable and it is impossible to predict the service life.

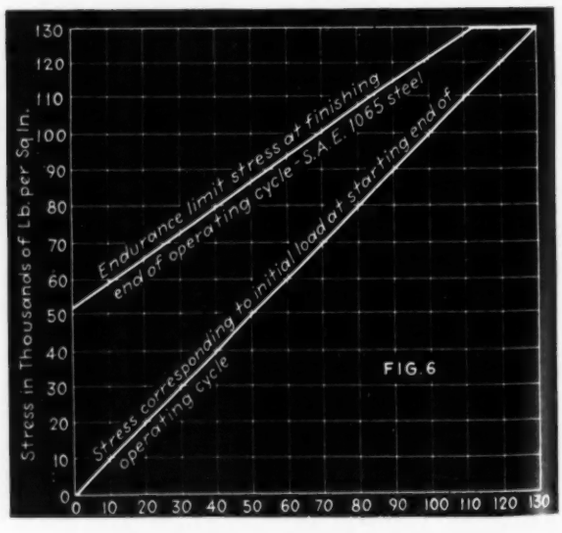
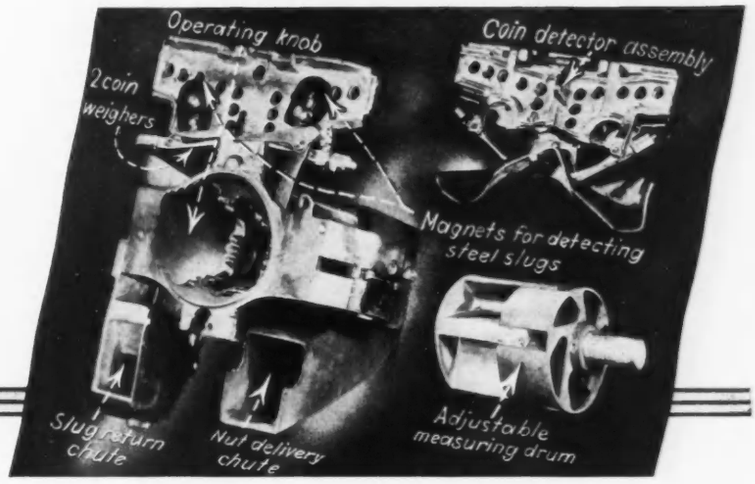


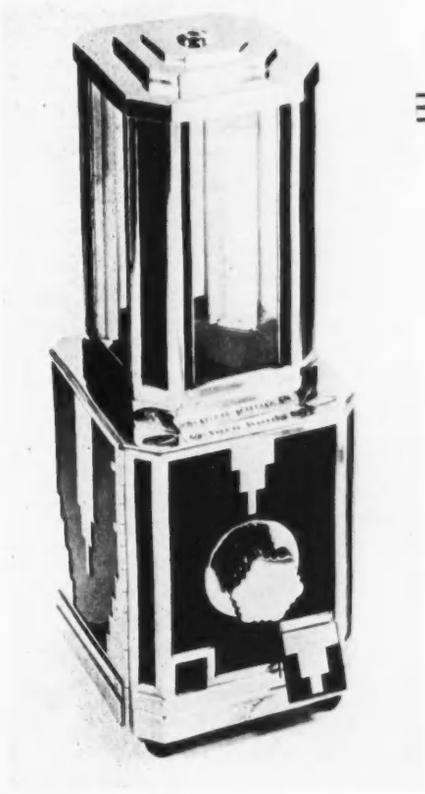

***Fig. 6—Chart based on tests of oil tempered carbon steel wire S.A.E. 1065 showing the limiting stress at end of operating cycle because of fatigue in relation to the initial stress set up by a preload at starting end of cycle***

# PRODUCT DEVELOPMENTS

## *Slug-Proof Mechanism for a Vending Machine*




***Zinc die-cast outer shell of nut vending machine is finished in chrome and baked enamel to resist atmospheric corrosion. Coin detecting and dispensing mechanism unit above is made almost entirely of zinc and aluminum die-castings***



• The volume of salted nuts, delivered by a coin, can be adjusted in the Stewart & McGuire vending machine. Also, two coin slots are provided, one for pennies and the other for nickels. Insertion of a nickel will cause the delivery of five times the amount delivered by a penny. In addition to this, the mechanism is so designed that mutilated coins and slugs are rejected.

After insertion, the coin travels through an accurately dimensioned chute having deflectors which shunt out coins that are of incorrect thickness or diameter. As the coin rolls down the chute, it travels past a permanent magnet and then strikes a balanced arm. The former, by magnetic attraction, deflects iron or steel slugs, while the latter causes coins of improper weight to be rejected. Any deflected coin falls into a compartment so arranged that upon attempting to turn the operating knob of the machine, the false coin is returned.

To make adjustable the volume of nuts delivered, the measuring drum, divided into five compartments, is made of two slidable members as shown in the illustration. By positioning one piece axially, the volume of the compartments are changed accordingly. Thus the amount of merchandise delivered can be varied but the volume for a nickel will always be five times that for a penny.

When in operation, the difference between the diameter of a cent and a nickel determines the volume of salted nuts that will be delivered. The smaller diameter of the cent allows a stop lever to rise only a short distance and then to drop into the stop position as the cent passes from under the lifting pin, allowing only one-fifth of a revolution of the measuring drum. The larger diameter of the nickel raises the stop lever high enough so that a projection thereon holds it open for a complete revolution of the drum. Since the delivery drum, mounted on the same shaft as the handle and turning with it, is divided into five compartments, the nickel will, therefore, cause the machine to deliver five times the amount delivered by a cent.

So that it could be manufactured economically, the mechanisms had to be simple and easy to assemble. To procure the required dimensional accuracies without expensive machining operations, die-castings are used. There are 24 die-castings in the assembly, of which 20 are of zinc and 4 of aluminum. The latter, which include the bottom of the nut container, measuring drum and chutes, are used wherever the salted nuts come in contact with the metal. They are Alumilited by the anodic process for protection against corrosion from the salt. Other parts such as bolts and nuts, lock washers, screw machine parts and stampings are protected by either cadmium or nickel plating. Springs are made of stainless steel so that neither the kind of merchandise nor exposure can cause their deterioration.

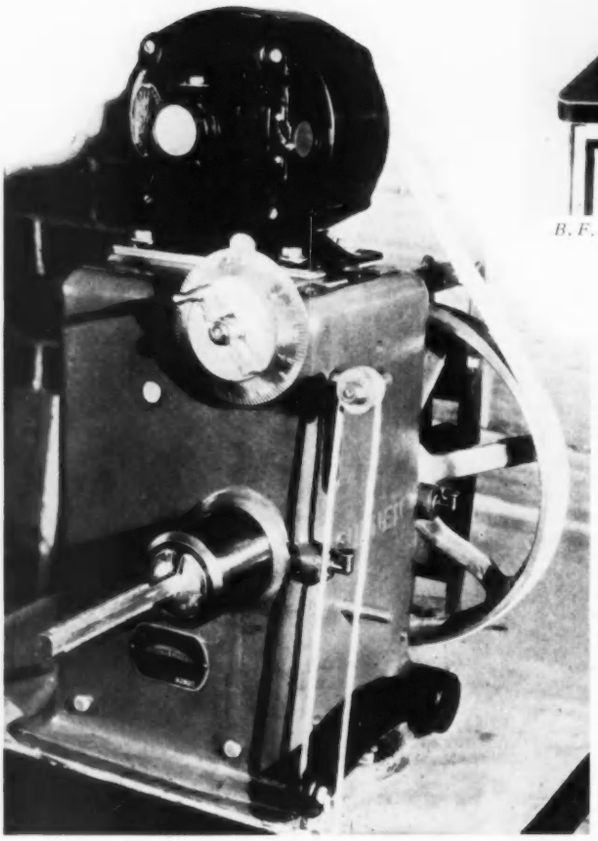
## •*Finishing Holes to 0.0001 In. Limits*

With the precision hone made by the Sunnen Products Company, unskilled labor can be used for grinding, honing or lapping holes within 0.0001 in. limits. Holes ranging from 0.480 in. to 2.400 in. diameter in such parts as air compressor cylin-

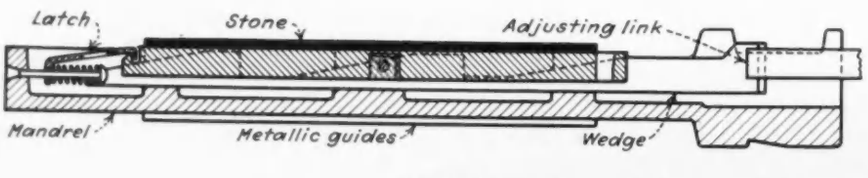
ders, hydraulic cylinders, bushings, ball bearing races, single and cluster gears, musical instrument valves, pistons, and other pieces made of any soft or hard metal, with the exception of lead or babbitt, can be finished on this machine. As shown in the accompanying sketch, a single stone is mounted on a slidable bar, the later being actuated by a three-step wedge. These are assembled into different sized mandrels having longitudinal two-point bearings in the work to be finished, the stone making a third bearing. It is possible to change quickly from one job to another.

The wedge for controlling the stone is actuated by a foot pedal and cord through a crank lever at the side of the machine as shown in the illustration. This crank lever imparts a longitudinal pull to the wedge through a system of links and a feed rod extending through the spindle, thereby raising the stone to contact the hole surface. The feed rod being linked to the mandrel wedge at the front, terminates at the opposite end in a thrust bearing fastened to the feed arm link. The upper end of the feed arm is used as a pivot when expanding the stone with the foot pedal, the position of the pivot being controlled through an adjustable screw. A dial, graduated to 0.0001 in. is mounted on the front end of the screw, through which a duplication of sizes can be maintained.

The foot pedal cord also runs over a pulley on the end of the idler pulley shaft control arm. Any increase in presure on the foot pedal increases the tension of the drive belt sufficiently to take care of the increased friction of the stone against the work. When the pressure is released from the foot pedal, the stone recedes automatically being actuated by a coil spring in the spindle. Pieces with small holes, up to 7 in. in length and without keyways can be held in the hand when finishing.



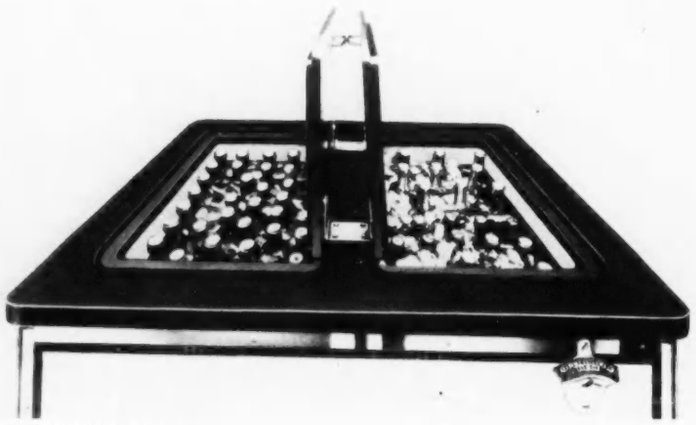
***Micrometer dial, graduated to 0.0001 in., controls adjustment of the hone in work. Stone is supported and raised by a 3-step wedge so mounted to assure parallelism with hole. Wedge recedes when foot pressure is released***




## •*Large Cover Molded of Hard Rubber*

The one-piece top of the Duo-Cold Cooler manufactured by the Eagle Foundry Company is believed to be the largest molded hard rubber product successfully produced for commercial use. This top with two hinged covers of the same material measures 38 in. long by 28 in. wide.

Hard rubber was adopted for this portion of the cabinet to avoid danger of chipping. It is said to be rust- and corrosion-proof and because there is no metal-to-metal contact, operation is quieter. Not only does the material have high insulating qualities, but the appearance is pleasing as well. Heretofore, this material could not be molded into large and accurate parts.

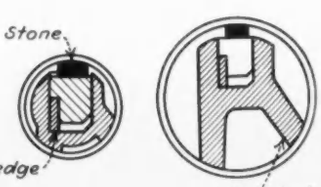


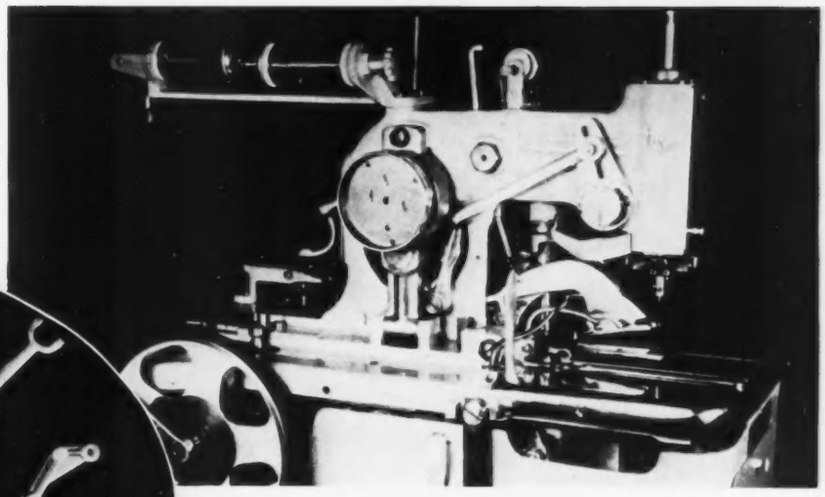
*B. F. Goodrich Company*

***Cover frame of bottle cooler is molded in one piece of hard rubber 38 in. long by 28 in. wide***

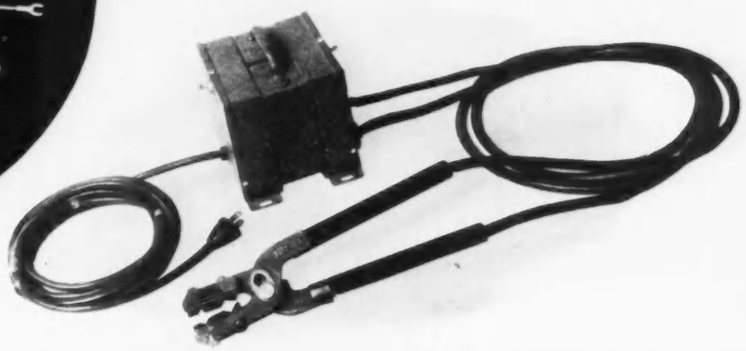
## •*Carbon Blocks Serve To Hold and Heat*

Renewable carbon blocks for the jaws, and insulated from the body of the pliers, serve not only as grippers but also as the electrodes of the "Thermo-Grip" pliers for soldering small parts. With reference to heating, their principle of operation is the same as that of resistance welders. An extension cord connects the carbon blocks to a transformer which when plugged in to a 110-volt a.c. supply delivers low-voltage high-amperage current when the pliers grip the work to be heated. As in





***One of the most delicate of sewing operations is that of making eyelet buttonholes as produced by this machine wherein aluminum alloy parts replacing those formerly made of ferrous metal, allow 50 per cent higher speed***



***In the "Thermo-Grip" pliers, the carbon jaw pieces hold the part and serve as the electrodes for "resistance-heating" the part to be soldered***

electric welders, heating begins instantaneously.

Used for soldering large or small fittings, wiring terminals and similar parts, this device holds the work and applies both heat and pressure simultaneously, thereby greatly facilitating the soldering operation. To make the pliers adaptable to soldering a range of sizes, the jaws have "two-position" openings and the transformer has a high and low heat control in addition to the on-off switch.

Other advantages of the "Thermo-Grip" pliers as compared to the conventional soldering torch are the absence of an open flame, better heat control, and less bulky to handle.

## •*Better Eyelets Through Higher Speed*

By making the reciprocating and oscillating parts of aluminum alloys instead of steel or malleable iron castings, it was possible to increase the speed of the Reece eyelet buttonhole machine 50 per cent. Not only were inertia forces lowered because of decreased weight, but the aluminum alloy parts are more durable as evidenced by less frequent replacements.

Commenting on the adoption of this material in their machines, Franklin A. Reece, president of the Reece Buttonhole Machine Company, states in part as follows: "We have found the use of Bohnalite to be a very worthwhile change. In spite of the higher speed, namely from 1,200 up to 1,800 reciprocations per min., whereby more uniform eyelets are being produced, the life of the machinery has been increased and the part replacements have diminished in number. The rotary cam in the head of the machine, through links and a rocking lever, imparts oscillation to the looper mechanism. Because of the nature of the machine and the stitch required, the motion of the looper carrier is necessarily abrupt and intermittent. Replacements of the cams and cam followers have dropped 80 per cent, which is attributed to the change in material.

"Because of inertia in starting and stopping the machine, the clutch dog which drives and also stops the machine, formerly made of a steel forging, has dropped 50 per cent in replacement. By changing from steel to aluminum alloy, replacement of the looper drag link has been reduced 90 per cent. A saving in weight of 42 per cent was made.

## •*Random Jottings About New Developments*

A molded plastic seal is used as the top cap of a flashlight battery. Not only do the seals serve as insulating disks, but they eliminate corrosion resulting from condensation, thereby prolonging life.

* * *

By substituting a fabricated structure of welded tubular steel in place of that made of structural steel, it was possible to build a boom 100 ft. long for use on a crawler crane. As the weight per foot was reduced from 125½ lb. to 75½ lb. it was possible for the machine to lift normal loads more than 80 ft. high without undue tipping strain.

* * *

Instead of resorting to glue as a binder in the manufacture of plywood, resins are now being used to cement the laminations together. It is said that not only is the new product weather-proof, but vermin-proof as well.

# *Spiral Diagrams to Solve Vibration Damping Problems*

DR. H. O. FUCHS
*Engineering Department, Delco Products Corporation*

**A method to analyze the effects of any kind of damping force such as exerted by automobile shock absorbers, landing gears, dash pots, electric dampers, rubber mounting and similar devices**

WITH the introduction of high speeds, there have been forced upon the attention of the engineer the problems of mechanical vibrations. In all such problems we deal with masses moving about an equilibrium position, (for example, the vertical movements of an automobile body); with forces tending to bring the mass towards the equilibrium position (restoring forces such as springs); with forces tending to move the mass away from the equilibrium position (exciting or unbalancing forces); and with forces resisting movement of the mass. These last are the damping forces such as exerted by friction, shock absorbers and other forms of vibration dampers.

For undamped vibrations and for two special cases of damping, solutions of the problem have been shown and explained in books on vibration problems. The two special cases are that of constant friction and that of a resistance which is proportional to the velocity of movement. However, actual damping seldom shows exactly these characteristics and often these special relations are not desirable.

The spiral damping diagram is a simple, direct and universal means to analyze the effect of any kind of resistance on the motion of a mass under the influence of a restoring force, starting from any given position with any velocity. The application of this method to the problem of a constant mass under the influence of a constant rate spring will first be explained. Modifications for the consideration of variable masses and variable rate restoring forces can be introduced.

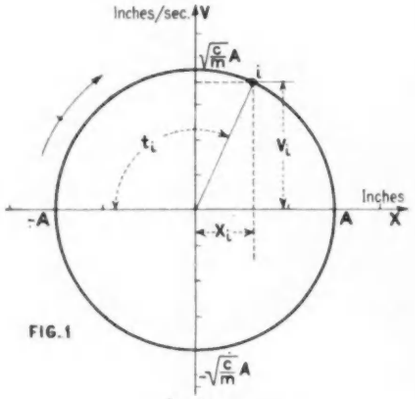



***Fig. 1—Typical diagram for undamped harmonic vibration***

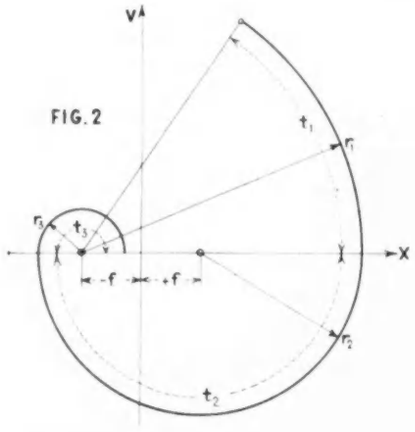



***Fig. 2—Spiral diagram for vibration damped by constant friction force***

### *The x-v Diagram for Undamped Vibration*

In Fig. 1 is shown the typical diagram for the harmonic movement of a mass. The velocities $v$ at any given instant $t$ are plotted over the displacements $x$ from the equilibrium position at the given instant. An undamped harmonic vibration, which is represented as a sine curve in the usual time-displacement diagram, appears in this $x$-$v$ diagram as a circle, if the scales are chosen in the right proportion, for reasons as follows:

If the maximum displacement (amplitude) is $A$, then,

$$x = At \sin \sqrt{c/m}$$

and $v = dx/dt$

$$= At \sqrt{c/m} \cos \sqrt{c/m}$$

where $c$ is the rate of the spring and $m$ the mass of the vibrating body. The maximum velocity is

$$v_{max} = A \sqrt{c/m}$$

If this maximum velocity is represented by the same length which represents the amplitude $A$, the graph of the vibration in the $x$-$v$ diagram becomes a circle of radius $A$ with the origin of coordinates as the center. This is because at any instant $i$ the abscissa will be the displacement $x_i$ which is equal to:

$$x_i = A \sin t_i.$$

Similarly, the velocity $v$ is the projection of the plotted point on the ordinate, which is equal to:

$$v_i = A \sqrt{c/m} \cos t_i.$$

The angle $t_i$ has taken the place of the angle $t \sqrt{c/m}$ of the classical equation. Therefore, we have a measure of the time in the $x$-$v$ diagram. To obtain the time in seconds between any two points on the circle, the angle between them must be divided by $\sqrt{c/m}$, the speed factor.

Another consideration leads to the same result. In an undamped free vibration the amount of energy contained in the system is constant. It oscillates between kinetic energy $1/2\ mv^2$ and potential energy $1/2\ cx^2$. Since no energy is dissipated by resistances, the sum of these two forms of energy must be

constant, and may be expressed by $cx^2 + mv^2 = 2 \times Energy = constant$ or $x^2 + (m/c)\, v^2 = constant$.

This equation describes an ellipse, which becomes a circle if the scale for velocities is chosen as $\sqrt{c/m}$ times the scale for displacements.

### *The x-v Diagram for Damped Vibration*

In addition to giving a concrete representation of the relations between displacements, velocities and energies in a free harmonic vibration, the $x$-$v$ diagram has practical advantages when damping forces or resistances enter into the problem. Thus, let us assume that the damping force at some instant is $F$. Instead of only the spring force $cx$ we now have the total force $-cx + F$. Measuring the damping, as is done frequently, by the displacement $f$ at which the spring alone would exert the force $F$, then:

$$F = -cf$$

The total force can then be written $-c(x + f)$. This means that, at this particular instant, everything goes on exactly as if the position of equilibrium had been shifted from the point $O$ to the point $-f$, and no damping force were present. The displacement from this point is equal to $x + f$ and the total force is equal to $-c(x + f)$, which is the total force with which we have to deal. Therefore at this instant, the point $-f$ can be used as the center of the arc of a circle which describes the relation of velocities and displacements. In words:

"The diagram of a damped vibration is composed of arcs of circles around those points on the x-axis at which the spring force is equal in amount with and opposite in direction to the instantaneous damping force."

The exact proof of this can be deduced from Proell's theorem, which states that in an $x$-$v$ diagram the subnormal is equal to the acceleration. Following are a few examples to show use of method.

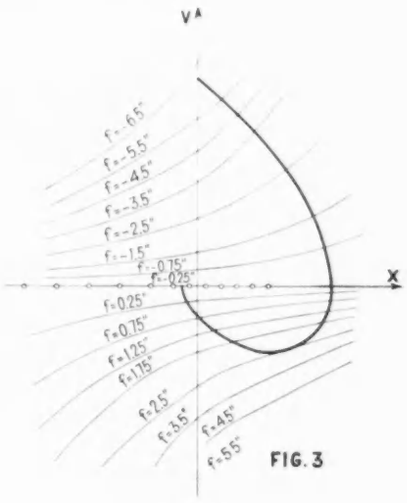
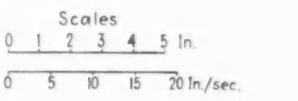



FIG. 3



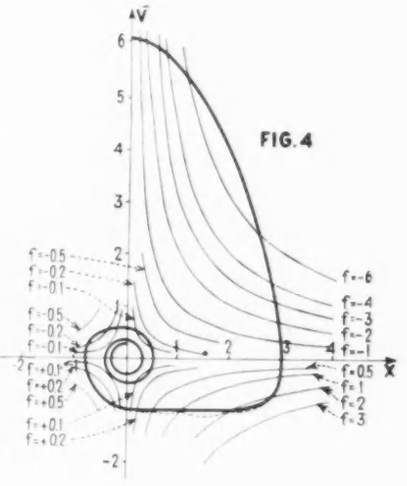




FIG. 4

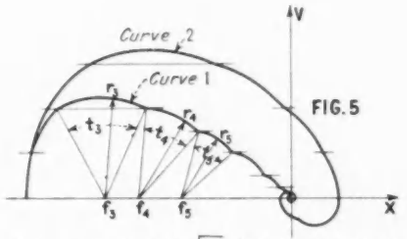



FIG. 5

For Curve 1: $f = V \times 2\sqrt{\frac{m}{c}}$ (critical damping)

For Curve 2: $f = V \times \sqrt{\frac{m}{c}}$ ($\frac{1}{2}$ critical damping)

***Fig. 3—Spiral diagram for vibration damped by a force, the magnitude of which is determined empirically***

***Fig. 4—Spiral diagram for vibration damped by a resistance proportional to velocity and displacement***

***Fig. 5—Spiral diagram for vibration damped by a resistance proportional to velocity***

### *EXAMPLE 1 — Vibration with constant friction.*

For a vibration with constant friction, the damping force is always equal in amount and changes direction when the direction of movement changes. As long as $v$ is positive, the damping force will be negative. This means that as long as the diagram is above the $x$ axis there will be arcs of circle around a point $-f$. As soon as the diagram line cuts the $x$-axis and enters the region of negative $v$, the center must be shifted to $+f$, as shown in Fig. 2. At the next crossing of the $x$-axis, the center must again be shifted and so on. A good rule is that the center must always be shifted in the direction in which the resisting force acts.

In Fig. 2 is shown how a vibration, starting with a given displacement and velocity and with a given constant friction, dies down after two and one-third half-oscillations. It also shows where the body will come to rest and how long it will take to do so. The time is found by adding the angles $t$ in radians and dividing their sum by the speed factor $\sqrt{c/m}$. Since circles about the origin are curves of constant energy content, the $x$-$v$ diagram also shows the amount of energy contained in the system at any instant. In the case shown in Fig. 2 for instance, a certain amount of energy remains even after the body comes to rest. This energy could be released by removing the friction.

### *EXAMPLE 2—The force F is given empirically.*

In Fig. 3 is shown the diagram for a vibration with a damping resistance which depends upon both position and velocity in a complicated manner as determined by experiment. The damping increases with the velocity and also with the distance from certain points, depending upon the direction of movement. The resistance is not proportional to the velocity or to this distance and never disappears completely. An analytical solution would be extremely difficult or impossible.

Here lines of constant damping force, as given by the experiments, are drawn and it is assumed that in the region between the lines the resistance does not vary. The movement starts with a given velocity at the position of equilibrium. The distances $f$ which must be used are inscribed in the different regions. Whenever a line of constant resistance is crossed, a new center is taken for the arc of circle. The centers are shown on the $x$-axis.

The error caused by the assumption of abruptly changing steps of resistance instead of the continuously variable resistance will be small if the change from one radius $r_i$ to the next radius $r_{i+1}$ is not too great. For $(r_{i+1} - r_i)/r_i = 40$ per cent, the error in time will be less than one per cent; for a radius increase of 100 per cent, the error will be less than 10 per cent.

### *EXAMPLE 3—Force F proportional to velocity and displacement.*

The action of a resistance which is proportional to both velocity and displacement is shown in Fig. 4. The procedure is the same as before. This figure shows how a certain limiting velocity and amplitude are reached before the end of the first swing. These limits cannot be exceeded, regardless of the amount of initial displacement and velocity. After the first swing, the damping rapidly decreases but it will never come to rest.

### *EXAMPLE 4—Force F proportional to velocity.*

The well-known case wherein the resistance is proportional to the velocity is shown in Fig. 5. Again zones of constant resistance are assumed but this time the zones are parallel to the $x$-axis. The motion represented by Curve *1* is not really a vibration because there is no second swing and the movement is aperiodic. Spiral damping diagrams are also valid for this type of movement. The vibration represented by Curve *2* will never come to rest, but succeeding amplitudes will always decrease in a certain proportion.

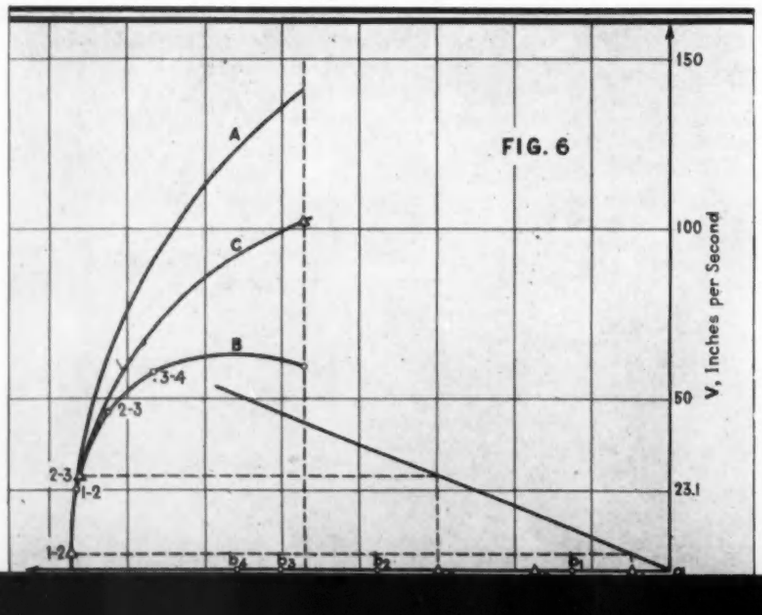
Incidentally, if $f$ is represented as a function of $v$ in this diagram, it will be a straight line through the origin. The tangent of the angle of this line with the $v$-axis is twice the ratio of the damping to the critical damping, which defines the boundary between aperiodic movements and continual vibrations.

### *EXAMPLE 5—An application to shock absorbers.*

In Fig. 6 are shown the details of execution of an example taken from an actual application of this method to the study of automobile shock absorbers.

Assume that the car be driven over a deep pothole 4 in. deep. The weight of the car, acting through the springs, will force the wheel down into the hole. The sooner it reaches ground, the better it will be for safety and tire wear. The question is: How long will it take the wheel to reach the ground, (*a*) without shock absorber; (*b*) with a shock absorber giving a resistance of 12 lbs. per in. per sec.; (*c*) with a shock absorber like (*b*), but modified by an upper limit of resistance (blow-off valve) of 330 lb. and a lower limit (static valve) of 55 lb.? The car specifications are: weight of the wheel 80 lb., load on the wheel 770 lb., spring rate 110 lb. per in., normal tire deflection 1 in. Assume that the car body move horizontally during this interval of time and that the tire bulge out instantly, so that 4 — 1 = 3 in. is the net drop of the wheel.

*Fig. 6 — Diagram for the solution of a problem in the application of shock absorbers*




For the diagram the calculations are as follows: The mass of the wheel $m = 80/386 = 0.207$ lb. sec.$^2$ in.$^{-1}$ Speed factor $\sqrt{c/m} = \sqrt{532} = 23.1$ radians per sec. The initial displacement of the wheel from its equilibrium position $x_o = (770 + 80)/110 = 7.73$ in.

Now draw as in Fig. 6 the $x$-axis and the $v$-axis on which latter 23.1 in. per sec. is represented by the length corresponding to 1 inch on the $x$-axis. Then mark off the starting point at 7.73 in. from the origin and the "ground" line at 7.73 — 3 = 4.73 in. from the origin. Finally lay off the line giving $f$ as a function of $v$ for case (*b*). It will be a line through the origin with a tangent of 110/(23.1x12). The diagrams can now be drawn and the time intervals calculated from them.

**Case (*a*).** The diagram is an arc of a circle around the origin from the point ($x = 7.73$ $v = 0$) to the ground line. It is found that: Radius $R = 7.73$ in. Chord $C = 6.81$ in. Ratio $C/R = 0.883$. Angle $t_a = 0.915$ radians. Time $T_a = 0.915/23.1 = 0.040$ sec.

**Case (*b*).** Assume the resistances to be constant in regions and use the centers shown by small circles and letters "*b*" on the $x$-axis

The following values are found or calculated:

| Region | v | f | R | C | C/R | $t_b$ |
|---|---|---|---|---|---|---|
| 1 | 0 to 23.1 | 1.26 | 6.47 | 1.01 | 0.156 | 0.156 |
| 2 | 23.1 46.2 | 3.78 | 4.00 | 1.08 | 0.270 | 0.271 |
| 3 | 46.2 57.75 | 5.04 | 3.00 | 1.04 | 0.347 | 0.349 |
| 4 | 57.75 up | 5.94 | 2.72 | 1.95 | 0.717 | 0.733 |

Sum of $t_b$ = 1.509 radians $T_b = 1.509/23.1 = 0.065$ sec.

The exact extent of region 4, where the velocity reaches a maximum, had to be found by trial and error, until the assumed $f$ corresponds to the average resistance given by the arc from this center.

**Case (*c*).** There are three regions to consider: one with constant resistance $f = 0.5$ up to $v = 4.6$ in./sec.; next with a variable resistance up to 330 lb. with corresponding $f = 3$ at $v = 27.5$ in./sec. and finally again a constant resistance of 330 lb. ($f = 3$). The different centers are indicated on the $x$-axis by triangles with the letter $c$. The following values are found:

| Region | v | f | R | C | C/R | $t_c$ |
|---|---|---|---|---|---|---|
| 1 | 0 to 4.6 | 0.5 | 7.23 | 1.94 | 0.268 | 0.269 |
| 2 | 4.6 27.5 | 1.75 | 5.99 | 0.73 | 0.123 | 0.123 |
| 3 | 27.5 up | 3.00 | 4.79 | 4.37 | 0.913 | 0.948 |

Sum of $t_c$ = 1.34 radians $T_c = 1.34/23.1 = 0.057$ sec.

The fact that the so called static resistance brings very little increase in time appears from the comparison of region $C_1$ with the arc of circle $A$. The advantage of the blow-off valve in the shock absorber appears not only from the comparison of the results, but also directly from inspection of the diagrams and estimation of the average speeds therefrom.

### *Widening the Scope of the Method*

Application of the spiral damping diagram to more complicated problems can easily be worked out. For self-excited and for resonant vibrations, negative damping must be used and the spirals will go away from the center instead of towards it. For variable masses and spring rates, the scale of $v$ must be changed in different regions, just as the centers $f$ were changed. Since the practical problems will always be new and different, and since the pith of a method can best be appreciated by working it out for oneself, the elaboration of variations is left to the reader and user.

# *Basis for Determining Manufacturing Tolerances*

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WALTER BARTKY
*Associate Professor, University of Chicago*

**Illustrating the use of the mathematical theory of probability in establishing permissible variations on individual parts based on a recognition and acceptance of actual manufacturing departures**

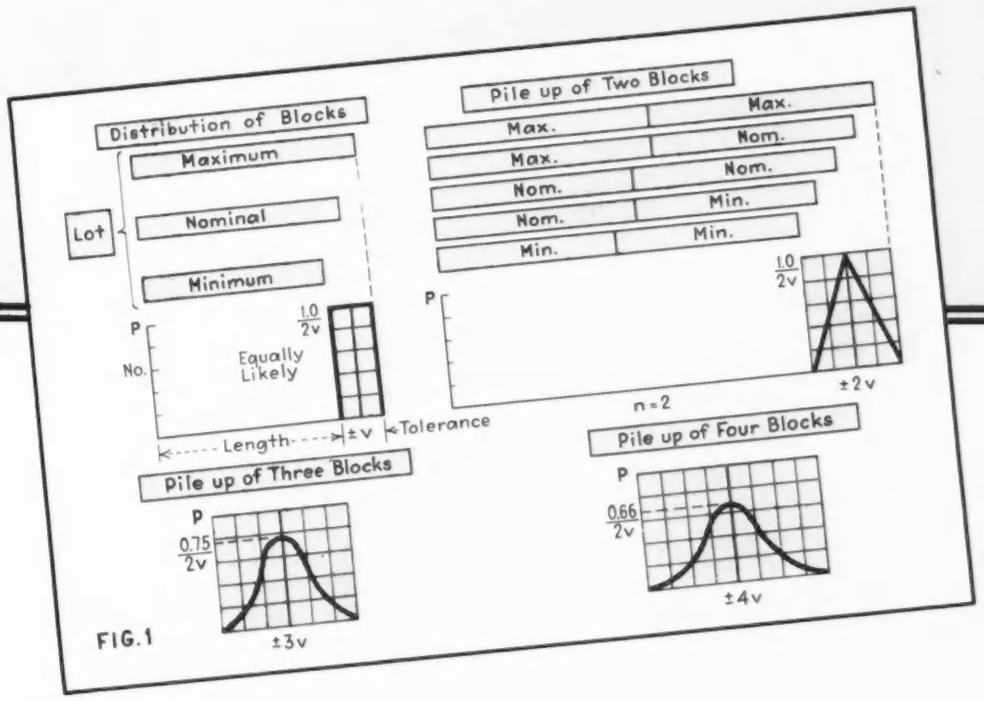



***Fig. 1 — Simplest illustration of mathematical theory of pile-up with blocks having the same tolerances***

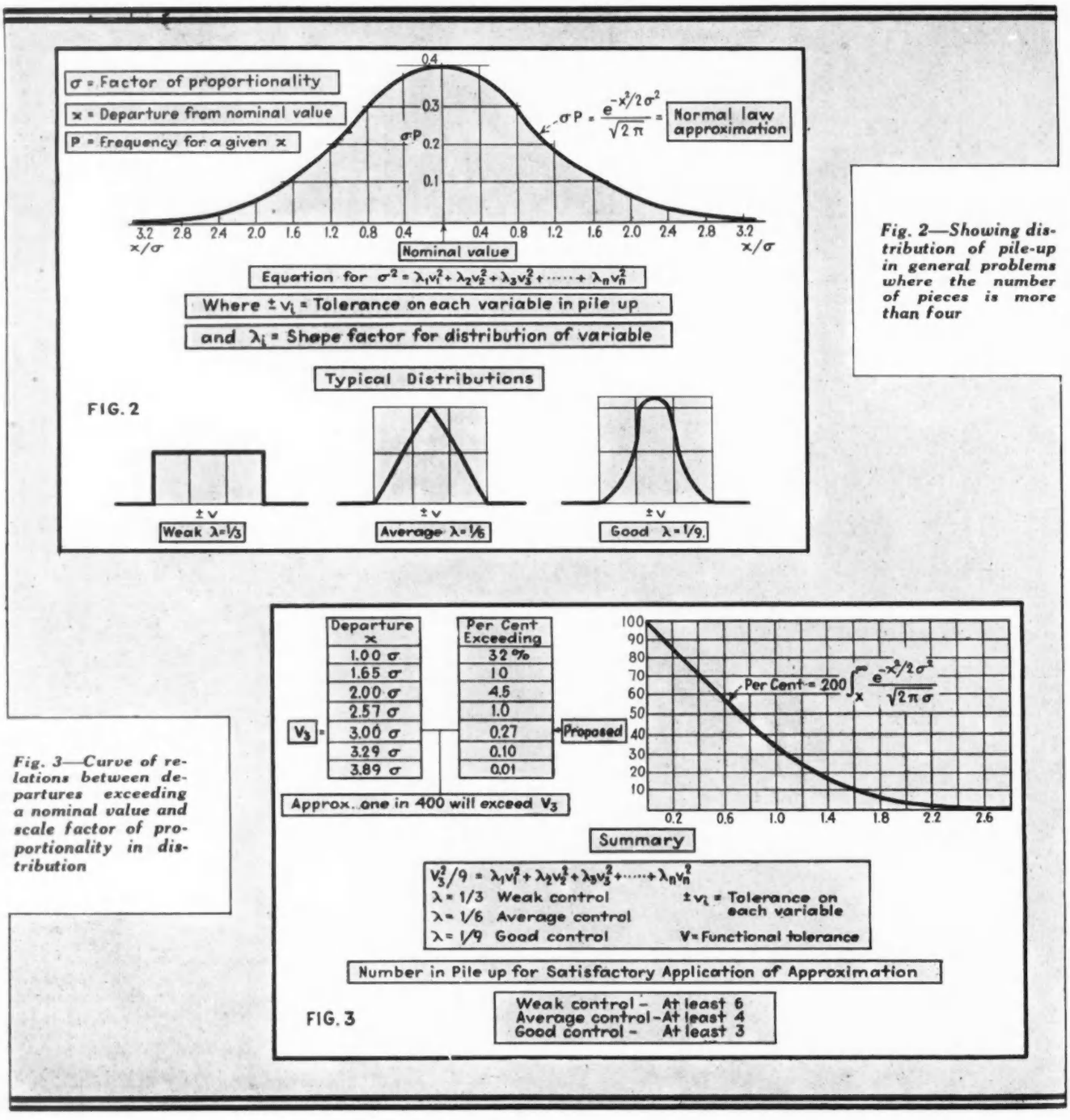
WHEN establishing manufacturing tolerances the tendency is to hold the product as close as possible to a fixed size or standard, with the result that the allowed variations specified on drawings are often exceeded. Once a design has been completed, the tolerances should be reviewed for their effect on the proper functioning of the finished product, and checked by the mathematical theory of pile-up for probable deviations.

To illustrate the mathematical theory of pile-up, take a lot of blocks that have been cut to some desired length but vary slightly from one block to the next. In this lot certain limits will be found; there will be a maximum length and a minimum length. Take the desired or nominal length to be midway between the maximum and minimum. Let the variation found from the nominal be called plus or minus $v$. This quantity will then correspond to the tolerance.

Assume that between the maximum and minimum lengths there are all intermediate lengths, or it is equally likely that the blocks are cut to any length within the tolerance. Technically this hypothesis is called a "Constant Probability Distribution." It places too high a value on the probability that the tolerance will be exceeded, so that if error occurs it is on the "safe side."

On Fig. 1 the distribution of the lengths in a lot of blocks is shown graphically. The ordinate represents the frequency $P$ with which the different lengths occur. If it is equally likely to have a length anywhere between the minimum and maximum, the distribution takes the form of a rectangle.

If two blocks of any length are put together the maxi-




mum variations obtained in the total length will be plus or minus $2v$ for all possible combinations. The extremes can be obtained in only one way. Two of the maximum blocks will give a maximum total length and two of the minimum blocks will give a minimum total length. A nominal total length, however, can be obtained by many different combinations. There will be a tendency in the pile-up for more lengths to be near the nominal value than near the extremes. The frequency distribution for the combined lengths will take the form of a triangle as shown in Fig 1. The area under the triangular curve is equal to the rectangle for the single blocks, the base being plus or minus $2v$ instead of plus or minus $v$.

For a pile-up of three blocks the maximum variation will be plus or minus $3v$. Again there will be a tendency for the lengths to cluster about a nominal value. The frequency of the distribution will take the form of a curve which is parabolic at the nominal value. For a pile-up of four blocks the curve will flatten out still further because of the increased spread of the extremes to plus or minus $4v$. As more and more blocks are added the curve becomes flatter and flatter and spreads out more widely at the extremes. It will be found that they are similar in shape and approach a certain limiting curve commonly called the normal law curve. For a pile-up of four or more blocks, the normal law curve may be used for determining frequency distribution of the final lengths.

A normal law curve is plotted on Fig. 2. The equation for the curve is shown in one of its common forms. The ordinate is proportional to the frequency $P$, and

the abscissa proportional to the departure $x$, in a pile-up from the nominal value. The scale factor $\sigma$, is introduced to keep the area under the curve equal to unity, it might be called the factor of proportionality upon which the distribution depends. For a pile-up of $n$ blocks it is found from the mathematical theory of probability that $\sigma$ is given by

$$\sigma^2 = n v^2/3, \tag{1}$$

where the frequency,

$$P = \left(\frac{}{e}\right)^{-x^2/2\sigma^2} \Big/ \sigma\sqrt{2\pi} \quad \text{approximately.} \tag{2}$$

Thus for a pile-up of 4 blocks,

$$P = 0.346\left(\frac{}{e}\right)^{-3x^2/8v^2} \Big/ v$$

For example at $x = 0$ this gives,

$$P = 0.346/v, \text{ approximately;}$$

but from chart No. 1 for four blocks $P$ was found to be $0.333/v$ exactly.

At $x = 4v$, normal law gives,

$$P = 0.346\, e^{-6}/v = 0.00086 \text{ approximately}$$

The exact value is of course zero. Hence, Equation (2) is a satisfactory approximation for a pile of four blocks. Furthermore, the greater the number in the pile-up the closer this approximation is to the exact value.

In the block problem it was assumed that the distributions were all of constant probability and with the same tolerances. The normal law approximation however, is not limited to this type of pile-up, but may be used for pile-ups with distributions other than constant probability and different tolerances. In this case the equation for $\sigma$ takes a more general form and is given by

$$\sigma^2 = \lambda_1 v_1^2 + \lambda_2 v_2^2 + \ldots\ldots\ldots\ldots \lambda_n v_n^2 \tag{3}$$

where

$\pm v_i$ = tolerance on a particular variable, or the $i$ th variable.
$n$ = number of variables involved in the pile-up.
$\lambda_i$ = a factor that depends upon the shape of the distribution for the $i$ th variable.

In Fig. 2 typical distributions have been plotted to show the shape they take and the nominal values for the factor given, corresponding to the three types illustrated. The mathematical derivation of these equations has been omitted from this article, but this should not interfere with an understanding of the properties of the equations or their application to every-day practical problems.

In the pile-up of blocks it was assumed that any length within the tolerances was equally likely to occur. For this type of distribution it is seen from Fig. 2 that $\lambda = 1/3$. It may happen that the variable groups itself close to the nominal giving distributions similar to the other two illustrated, with $\lambda = 1/6$ and $\lambda = 1/9$ respectively. The greater the tendency to cluster about the nominal value the smaller the factor $\lambda$. Consequently, a small value of $\lambda$ indicates high operational control. For practical application of these equations it is convenient to classify the different types of distributions into one of three as given in Fig. 2. It should be noted that for $\lambda = 1/3$ (weak), 33.3 per cent group about the nominal, for $\lambda = 1/6$ (average), 55.5 per cent and for $\lambda = 1/9$ (good), 66.6 per cent group at the nominal. The selection of which of these to use for a given variable depends upon the method or kind of operation employed in producing the parts. The most desirable way to determine which type to use is to actually measure a sample of the production, plot the frequency of the variable and compare the resulting distribution with those illustrated in Fig. 2. When this is impractical and no other knowledge of the distribution is available, a safe procedure is to take $\lambda = 1/3$.

To facilitate the application of Equation (3) to practical problems in which the final functional tolerances and the allowable departures or rejections are the important considerations in setting the tolerances on the individual parts; a cumulative curve for the normal law has been plotted on Fig. 3. The ordinate gives the per cent of the product exceeding a departure $x$ from the nominal value and as before the scale factor $\sigma$ is introduced in the abscissa to make it applicable to any pile-up. Given the allowable rejections or per cent exceeding and the final tolerance or departure $x$ is determined in terms of $\sigma$. For small percentages the table of departures given in Fig. 3 is more satisfactory to use than the curve.

For example, assume that if the final rejections of a product do not exceed one in every 400, or 0.27 per cent the situation is satisfactory. From the table it is seen that for 0.27 per cent, the tolerance on the pile-up should be set at plus or minus 3 $\sigma$, thus

$$V_3 = \pm 3\sigma \tag{4}$$

where $V_3$ is the final tolerance.

On combining Equation (3) with (4) a relation connecting the final functional tolerance with the tolerances on the parts for a pile up of $n$ is obtained,

$$V_3^2/9 = \lambda_1 v_1^2 + \lambda_2 v_2^2 + \ldots\ldots\ldots \lambda_n v_n^2 \tag{5}$$

These equations are only approximations, the greater the number in the pile-up the better the approximations. The minimum number in the pile-up depends partly on the kind of control, or the value $\lambda$. It is recommended that if the control is "weak," there should be at least six in the pile-up, for "average" at least four, and for "good" at least three.

In a broad sense there are two types of problems. First, the determination of the functional tolerance given the tolerances on the parts. For example, if the $v_i$ are all equal and there is average control, $\lambda_i = 1/6$ then,

$$V_3^2/9 = (v_1^2 + v_2^2 + \ldots\ldots\ldots v_n^2)/6, \text{ or } V_3 = 1.22v\sqrt{n}$$

It is obvious that the maximum deviation would be the additive sum of the $v_i$,

$$v_1 + v_2 + \ldots\ldots\ldots v_n = nv$$

The tolerance $V_3$ from a probability standpoint varies as the square root of the number $n$ in the pile up while the maximum varies directly with $n$. A numerical example is shown in Fig. 4.

The inverse problem is given the functional tolerance $V_3$ to set the tolerances on the parts. In practice it is usually found that certain of the $v_i$ are fixed and the number that is subject to change is limited. Suppose that only two tolerances are subject to change. Thus assume that in the following equation,

$$V_3^2/9 = \lambda_1 v_1^2 + \lambda_2 v_2^2 + \ldots\ldots\ldots\ldots \lambda_n v_n^2,$$

only $v_1$ and $v_2$ may be varied. The equation then determines the sum,

$$S = \lambda_1 v_1^2 + \lambda_2 v_2^2.$$

Consequently, the value of $v_1$ depends on $v_2$, $v_1$ may be increased by decreasing $v_2$ or vice versa. The procedure to follow depends on the cost.

The previous problem of pile up was one in which

**To Determine Functional Tolerance $V_3$**

$v_i$ All equal to v - Normal control $\lambda_i = \frac{1}{6}$

$\frac{V_3^2}{9} = \frac{1}{6}(v_1^2 + v_2^2 + \cdots + v_n^2)$ or $V_3 = 1.22\sqrt{n} \times v$

If $v = \pm 0.01''$ and $n = 8$, then $V_3 = \pm 0.0346''$

Maximum deviation $= \pm(v_1 + v_2 + \cdots + v_n) = \pm 0.08''$

If $n = 4$, then $V_3 = 1.22\sqrt{4} \times 0.01 = \pm 0.0244''$

while maximum deviation $= \pm 0.04''$

**Inverse Problem - Functional Tolerance Fixed by Performance**

$\frac{V_3^2}{9} = \lambda_1 v_1^2 + \lambda_2 v_2^2 + (\lambda_3 v_3^2 + \cdots + \lambda_n v_n^2)$

Assigned reducible fixed by practice

therefore $\lambda_1 v_1^2 + \lambda_2 v_2^2 =$ Difference

i.e. $v_1$ may be increased by decreasing $v_2$

$v_2$ may be increased by decreasing $v_1$

The procedure to follow is determined by cost

***Fig. 4—Example of procedure followed in solving the two general types of problems***

the variables were additive. The method may be generalized to include cases where the dependent variable to be controlled is expressable mathematically in terms of the independent variables arising in the process of manufacture. Let the dependent variable be $f$ and the independent variable be $x_1, x_2, \ldots x_n$; and assume that the relation between the dependent variable and the independent variables is known, as

$$f = f(x_1, x_2, \ldots \ldots x_n)$$

If $V_3$ is the tolerance for $f$ then

$$V_3^2/9 = \lambda_1 \left(\frac{\delta f}{\delta x_1}\right)^2 v_1^2 + \lambda_2 \left(\frac{\delta f}{\delta x_2}\right)^2 v_2^2 + \ldots \lambda_n \left(\frac{\delta f}{\delta x_n}\right)^2 v_n^2$$

whence the partial derivatives are evaluated at the nominal values. As before $v_1, v_2, \ldots v_n$ are tolerances on the independent variables while $\lambda_1, \lambda_2, \ldots \lambda_n$, are factors depending on the type of distribution and the $V_3$ is such that about one in four hundred will exceed this tolerance. It will be noted that for $f = x_1 + x_2 + \ldots + x_n$

$$\frac{\delta f}{\delta x_1} = 1$$

and Equation 6 reduces to Equation 5.

The maximum possible deviation is given approximately by

$$\text{Max. dev.} = \left|\frac{\delta f}{\delta x_1}\right| V_1 + \left|\frac{\delta f}{\delta x_2}\right| V_2 + \ldots \ldots + \left|\frac{\delta f}{\delta x_n}\right| V_n \qquad (7)$$

where the absolute value of the partial derivatives are taken at the nominal value.

Whence the relation between tolerance is by Equation 6.

$$\left(\frac{V_f}{f}\right)^2 = 9\left[\lambda_N \left(\frac{V_N}{N}\right)^2 + 9\lambda_D \left(\frac{V_D}{D}\right)^2 + 16\,\lambda_d \left(\frac{V_d}{d}\right)^2 + \lambda_g \left(\frac{V_G}{G}\right)^2\right]$$

whence subscripts denote the variables. The maximum deviation in $f$ is by Equation (7).

$$\text{Max. dev. in } f = f\left[\frac{V_N}{N} + \frac{3V_D}{D} + \frac{4V_d}{d} + \frac{V_G}{G}\right]$$

The variables are here taken at their nominal values. It will be noted that expressions like

$$100\,\frac{V_N}{N}, \text{ and } 100\,\frac{V_D}{D}$$

are the tolerances expressed in per cent.

As an illustration of the application to non-additive variables, consider the problem of controlling the flexibility of a coiled spring. The compression per lb.

$$f = 8\,N\,D^3/G\,d^4$$

where

$N$ = number of effective turns
$D$ = mean diameter of coil
$d$ = diameter of wire
$G$ = torsional modulus of elasticity

and the tolerances on $N$, $D$, $d$, and $G$ are $\pm 5$, $\pm 1.5$, $\pm 0.8$ and 3 per cent respectively, the distribution was taken as good for all the variables, or

$$\lambda_N = \lambda_d = \lambda_D = \lambda_G = 1/6$$

The partial derivatives are

$$\frac{\delta f}{\delta N} = \frac{8D^3}{Gd^4} = \frac{f}{N}$$

$$\frac{\delta f}{\delta D} = \frac{24ND^2}{Gd^4} = \frac{3f}{D}$$

$$\frac{\delta f}{\delta d} = \frac{32ND^3}{Gd^5} = \frac{4f}{d}$$

$$\frac{\delta f}{\delta G} = \frac{8ND^3}{G^2d^4} = \frac{f}{G}$$

Consequently,

$$\left(\frac{V_f}{f}\right)^2 = \frac{3}{2}\left[(0.05)^2 + 9(0.015)^2 + 16(0.008)^2 + (0.03)^2\right]$$

$$\frac{V_f}{f} = 0.098$$

that is the tolerance in $f$ is a little less than $\pm$ 10 per cent.

The maximum deviation is given by $f = 0.05 + 3\,(0.015) + 4\,(0.008) + 0.03 = 0.157$ or a little less than plus or minus 16 per cent. Under the hypothesis involved only about one in 400 will be outside the tolerance $V_f$, and only 100 will deviate from the nominal value by more than 10 per cent of $f$. Not any should deviate by more than 16 per cent if the tolerances are held.

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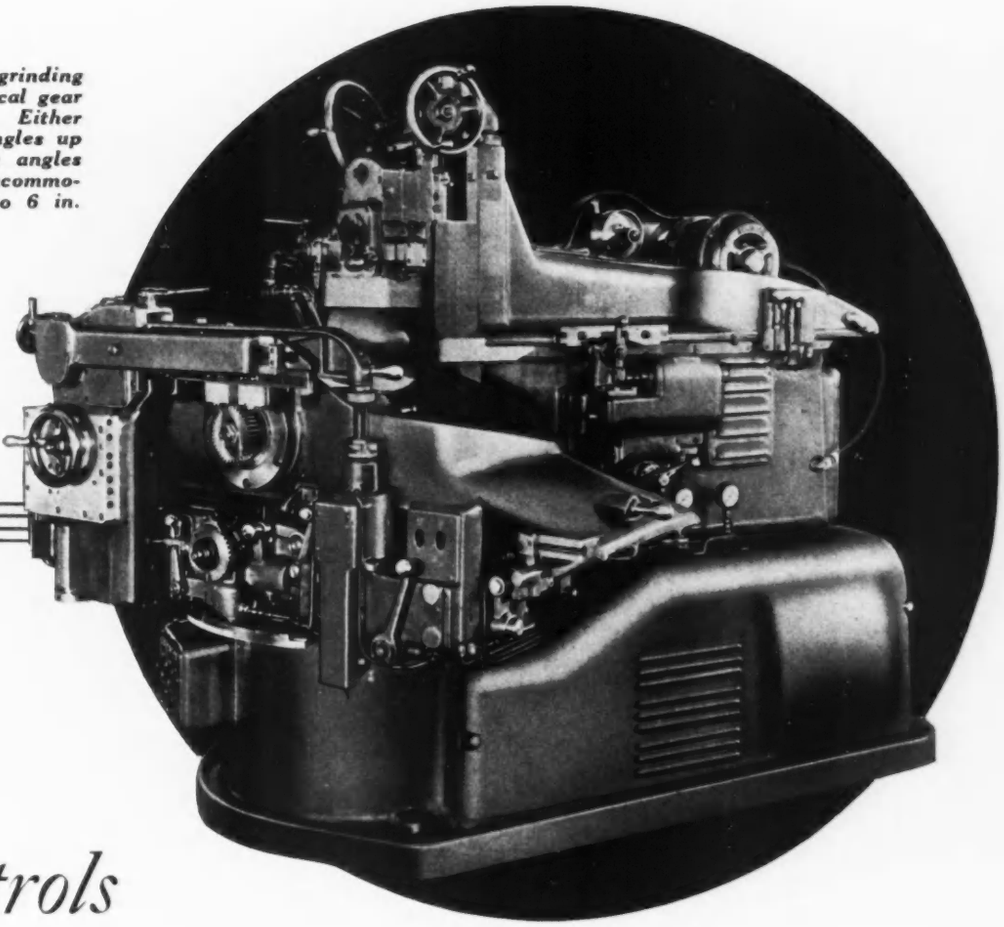


***FIG. 1—Reciprocating grinding wheel finishes spur or helical gear teeth to within 0.0002 in. Either right or left-hand helix angles up to 45 deg. and pressure angles from 13 to 30 deg. are accommodated. Tooth faces up to 6 in. can be ground***

# *Circuits and Controls for a Hydraulic-Electric Gear Grinder*

**Hydraulic control as applied to gear grinder for involute spur and helical teeth wherein the wheel spindle ram, table, indexing, automatic knockoff and oil filter are actuated by fluid power**

IN THE hydraulically operated 10 in. spur and helical gear tooth grinder manufactured by the Pratt & Whitney Company, shown in Fig. 1, both sides of one tooth space are generated simultaneously. A grinding wheel is mounted on a reciprocating horizontal ram, which is somewhat similar to that of a horizontal shaper. The grinding wheel is trued with straight tapering sides and face to correspond to the shape of a tooth in the master rack. Teeth being ground are rolled past the reciprocating wheel, guided by means of a master gear rolling on the master rack. When the work has rolled to a point where it is out of engagement with the grinding wheel, it is automatically indexed to the next tooth space. After all the teeth have been ground the machine is stopped automatically.

Two separate but interlocked hydraulic circuits with a common pump and reservoir are employed. As shown in Fig. 2, one actuates the table and indexing mechanism, the other reciprocates the wheel.

In the operation of the ram or wheel reciprocating circuit, oil in going from the pump to the ram cylinder passes through the ram speed control valve, the ram automatic stop valve and the ram reverse valve. The speed control valve is used only for manual regulation of the ram speed when setting up the machine. It is always fully open during normal operation, at which time the speed of ram reciprocation is regulated by the adjustable relief valve incorporated in the speed control valve. The automatic stop valve is held open by the pressure in the table circuit. As soon as the table motion is stopped, the pressure in its hydraulic circuit drops. This causes the automatic stop valve to close and thereby shut off the oil supply to the ram cylinder.

In Fig. 2 the valve positions are shown for moving the wheel to the right. At the end of the stroke the ram pilot valve is moved to the right from its central or

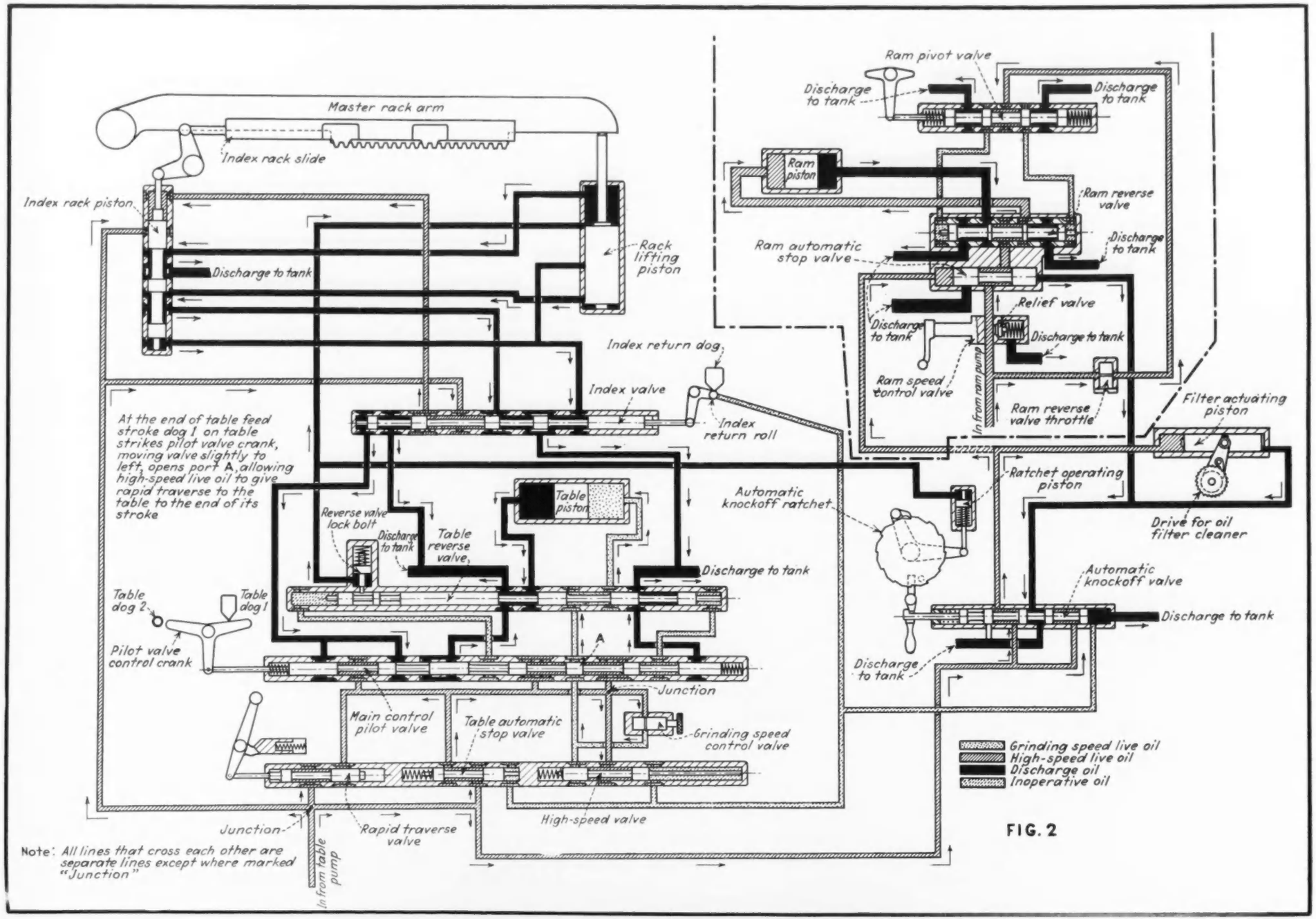



FIG. 2

**FIG. 2—Diagram shows all valves in positions occupied while grinding wheel is passing through tooth space of gear being ground the index valve being set for the next index**

neutral position by an adjustable dog on the ram, which contacts the lever shown at the left of the pilot valve. Oil then flows through the ram reverse valve throttle and the ram pilot valve to the right-hand end of the ram reverse valve, moving the latter valve to the left. The oil at the left of the valve discharges through the ram pilot valve and then to the tank.

In the operation of the table and indexing circuit, the oil is pumped to the table cylinder through the table automatic stop valve, grinding speed control valve, main control pilot valve and the table reverse valve to the right-hand end of the table cylinder piston, thereby moving the table to the left.

At the end of the table feed stroke to the left, its movement speeded, as indicated in Fig. 2, by table dog 1. At the end of the stroke dog 2 moves the pilot valve control crank, moving the valve to its extreme left position. Oil then passes to the right hand end of the table reverse valve, moving this valve to the left, to a neutral position, further movement being prevented by the reverse valve lock bolt. Thus the table is locked in a fixed position while the work is being indexed.

Indexing is performed when oil passes to the left end of the index valve, moving the index return roll upwards, the dog being out of contact at this time. A supply of oil then enters at the lower end of the rack lifting piston, moving the piston upwards, and thereby lifting the master rack out of mesh with the master gear, the oil from the top of this cylinder discharging to the reservoir. At this time oil flows to the lower end of the index rack piston, moving the index rack slide to the right thereby indexing the master gear.

*FIG. 2—Diagram shows all valves in positions occupied while grinding wheel is passing through tooth space of gear being ground the index valve being set for the next index*

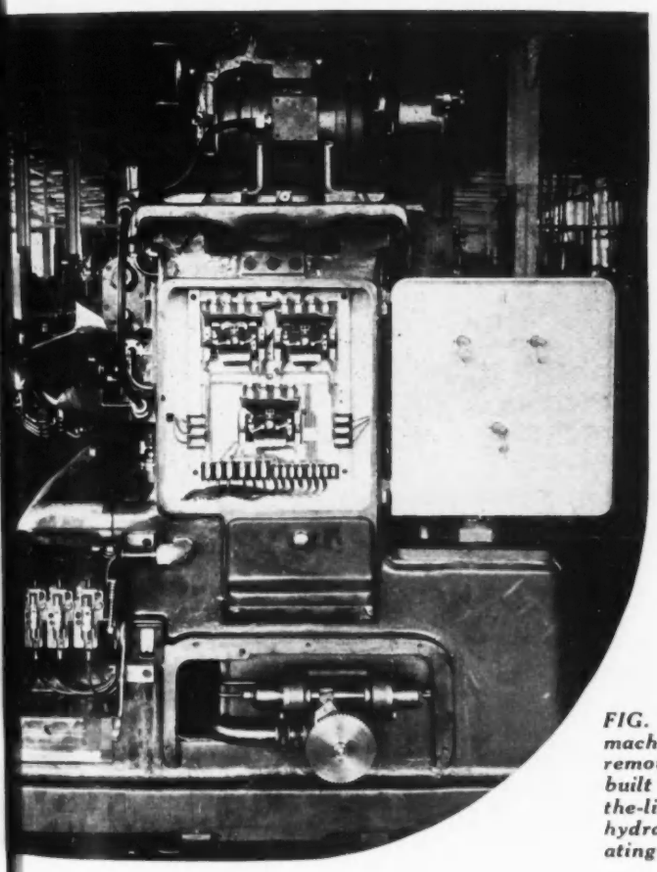


***FIG. 3—Rear view of machine, with covers removed, showing in-built solenoid, across-the-line starters and hydraulic oil filter actuating mechanism***

At top stroke of the index rack piston, a port is opened, forcing oil to the upper end of the rack lifting piston and thereby moving it downwards. In its lowermost position, a port is uncovered which allows oil to pass to the ratchet operating piston thereby moving the automatic knockoff ratchet wheel an amount corresponding to the indexing. Simultaneously, the oil lifts the reverse valve lock bolt allowing the table reverse valve to move to its extreme left-hand position. Oil then flows through the main control pilot valve to the left-hand end of the table piston cylinder moving the table rapidly to the right or to the grinding position. The oil passes through the grinding speed control valve, the table continuing its movement to the right at reduced grinding speed.

As the table passes through the central portion of its movement, the index return dog depresses the roll, moving the index valve to the left. This causes the indexing rack piston to move downwards, moving the index rack slide to the left against an adjustable stop. Downward movement of the index rack piston to restore the index slide to its initial position, takes place when the index slide rack is out of contact with the master gear. Movement of the index rack slide is limited to one circular pitch of the master rack. The table continues to move to the right and the cycle of reversing and indexing is repeated. When the last indexing is completed a pin on the automatic knockoff ratchet wheel trips the latch that holds the automatic knockoff valve to the left, allowing the spring to move it to the right. The ram automatic stop valve is then moved to the left, stopping the ram reciprocation. To again put both hydraulic circuits into operation, the automatic knockoff valve is moved manually to the left until the latch holds it.

To operate the Cuno self-cleaning oil filter shown at the right side of the diagram, oil under pressure reciprocates the "filter actuating piston." This drives the connecting arm which carries a pawl that engages the ratchet wheel. Thus for each complete cycle of the reciprocating grinding wheel the cleaner is rotated a fraction of a revolution.

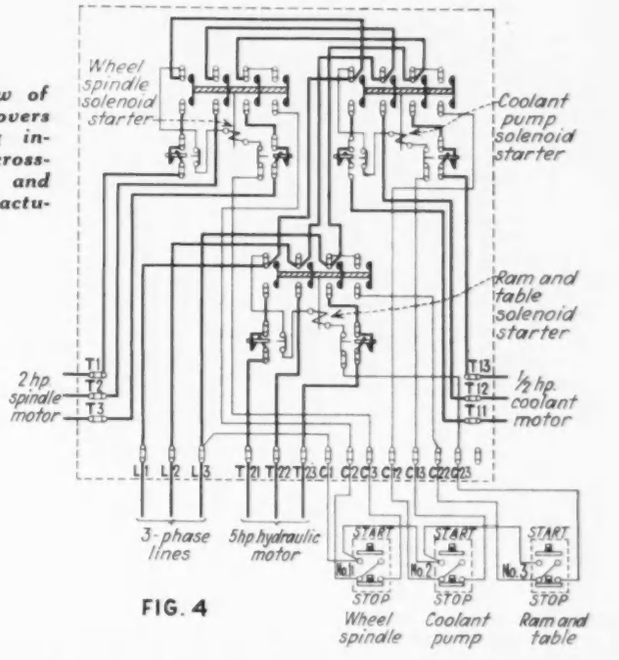



FIG. 4

***FIG. 4—Wiring diagram of gear tooth grinder as shown built-in at rear of machine in Fig. 3***

# *Using Same Parts for All Models Cuts Costs—Widens Markets*

## Analysis of field difficulties, pooling of engineering talent, keeping the best elements of previous designs, and consideration of shop operations all have a part in developing new air-conditioning units

THE new line of condensing units for air conditioning, recently announced by Westinghouse Electric & Manufacturing Company, illustrates the progress in design that can be achieved through a systematic analysis of the market and co-ordination of available resources. Convinced that the development of the mass market for small installations depends largely on supplying relatively trouble-free equipment, the objective was to develop a line of machines free of operating difficulties and entirely automatic. Patents and the experience of air-conditioning and refrigeration design engineers were pooled to reach that goal.

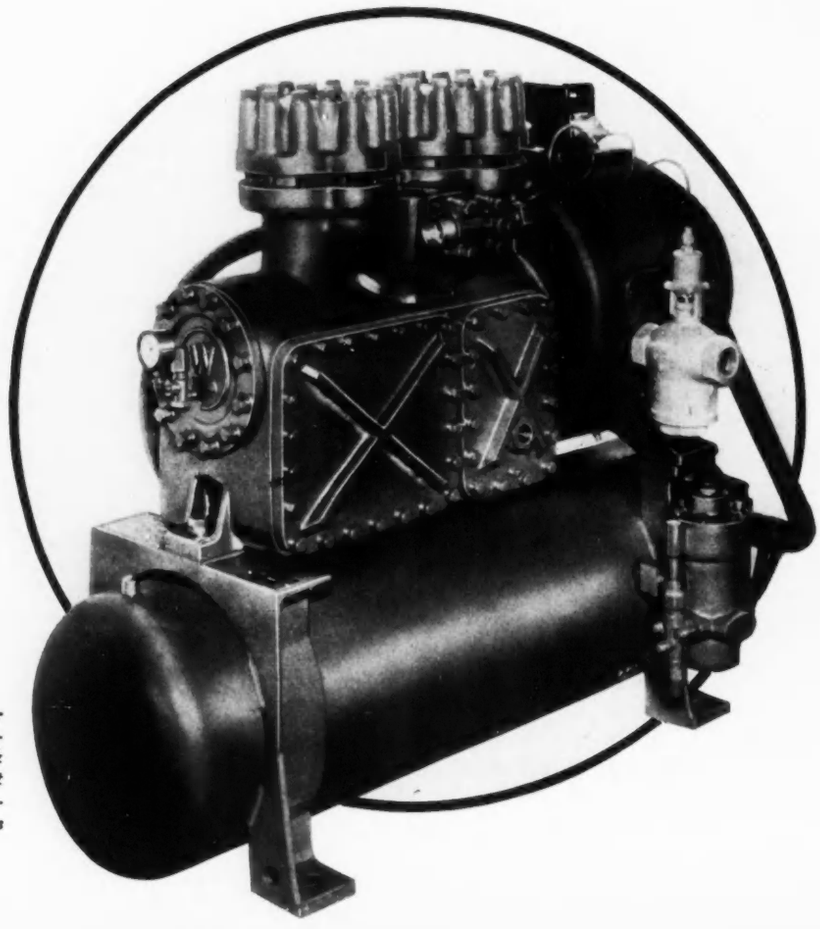
Before beginning the design of the new compressor much consideration was given to records of previous installations. A comprehensive survey indicated that most of the trouble experienced was caused by leakage of refrigerant at the driving-shaft seal of the compressor. Correction of this fault was not simple for the surfaces in contact must be perfectly machined and fitted at the factory and must maintain a tight fit throughout the life of the machine. Years ago the domestic refrigerator manufacturers solved this problem by placing the motor inside the compressor housing, thus doing away with the moving seal surfaces of a stuffing box. It was decided, therefore, to put the motor inside of the housing, thereby avoiding the chief source of trouble experienced with earlier designs.

The a.c. motor-driven compressors are completely inclosed and provided with adequate initial lubrication to assure long life. For those installations, about 5 per cent of the total, requiring the use of a d.c. motor, the open type with a seal is still used since the motor commutator cannot be inclosed in the refrigerant gas.

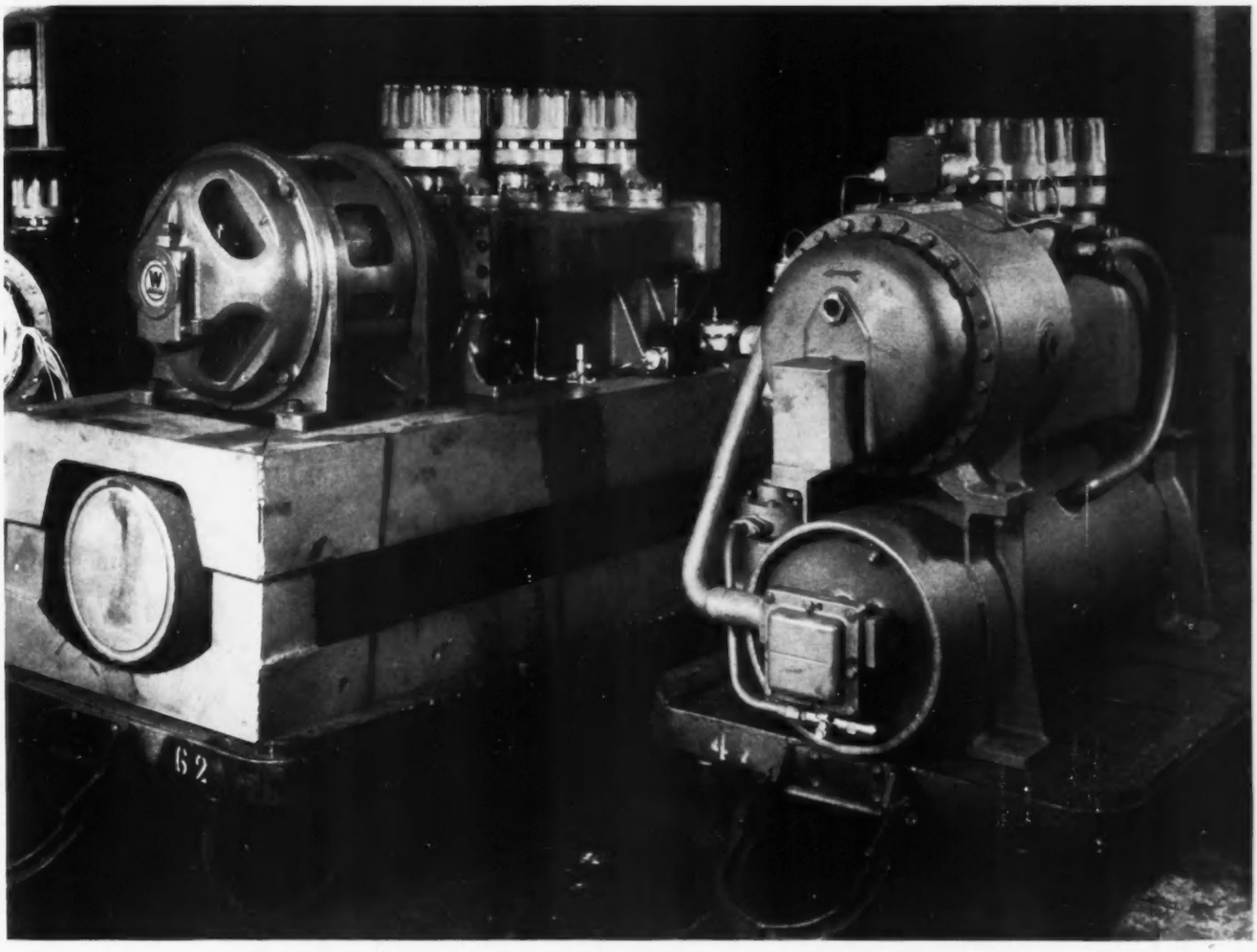
The motor rotor is mounted on and keyed to an overhung end of the crankshaft thus eliminating a coupling, an outboard bearing, and the necessity for lubricating the motor.

Cast integrally with the main frame are the cylinders, crankcase, water-jacketed motor housing, intake and exhaust manifolds. It is a difficult piece to cast since the cylinder walls must be smooth and the entire casting must be gas tight. The only piping is the discharge line to the condenser and the liquid line connection to the strainer.

With constant circulation of water through the cored shell surrounding the stator, the machine can be installed in places where there is little ventilation. Preliminary tests show that the unit can be completely boxed and run with safety.



***New type condensing unit for air-conditioning installations developed by Westinghouse Electric & Manufacturing Company to meet the demand for an automatic machine free of operating difficulties***



***New 4-cylinder sealless design shown at right has 94 per cent of the capacity and has less than one-half the weight of old style 6-cylinder machine at left***

A Micarta gear driven from the crankshaft drives a gear type oil pump. The pump can be readily taken out the side of the crankcase after removing the bolts in the end bearing. The lubricant is distributed to every part of the unit and is cooled through contact with the water-cooled motor housing. Oil pumped with the refrigerant is removed by a separator and returns to the sump for recirculation.

Parallel feed of oil to the bearings, rather than series feed, is used. In previous machines the oil flowed through the crankshaft some distance before it reached the main bearings. In the new design, strained oil from the pump is piped directly to each main bearing where it is distributed to the nearest connecting rods through holes in the crankshaft.

## *Valve Port Area Increased*

The valves, while of the same general design as formerly used have been improved by increasing the port areas considerably, with easier and less restricted passageways. Also the two plates forming the valve seat and housing are now aligned by a small pin which enters and fits a hole drilled in the cylinder head, thus making it impossible to assemble the valves other than in the correct position.

The cylinder is inherently balanced to counteract rotational forces that tend to vibrate the machine. However, twice in each revolution there is a secondary reciprocating force that tends to lift the machine off the base. To offset this the oil pump carries a harmonic balancer consisting of two small weights rotating opposite to each other at twice the crankshaft speed.

Similar to previous designs the condenser is of the shell and coil type, with the water passing through the coil, except for one marked difference. The new design offers the choice of two different pressure drops through the condenser coils for the same water flow, which is accomplished by a water manifold cover external to the condenser. These covers are shipped assembled so that the water flows through the two coils in series to suit ordinary city water pressures of 35 lb. per sq.in. or higher. If it is desired to pass more water through the condenser the two coils can be placed in parallel by removing the cover, dropping out a thin plate and replacing the cover.

The pressure drop will then be approximately one-eighth that of the series connection. The condensers are all copper and steel. No brass is used which can deteriorate by dezincification with certain types of water.

The new design is equipped with a pilot-operated water-regulating valve the power element of which consists of a bellows instead of diaphragm.

No mechanism of the size of air-conditioning units can be completely free of the occasional necessity for mechanical adjustments. Therefore, to make servicing

easy, the entire operating mechanism can be exposed by removing the side plates from the crankcase casting, without disconnecting refrigerant or water lines nor disturbing the unit itself.

After the requirements for a good machine were settled, the next problem was to decide how to make the machines of various sizes and build them at a reasonable cost. Since the number of machines of any one size to be built is relatively small, it was decided to build the various models with as many common parts as possible.

To accomplish the common parts idea, the three most popular sizes of machines were selected and compressors designed for 2, 4 and 6 cylinders with connected motor horsepower respectively of 7½, 15, and 25 hp. In this way the same parts are used for end bearings, pistons, piston rings, connecting rod, wrist pin, center bearings, cylinder heads, side covers, valves, oil pump and various condenser parts. The only major parts that differ for the three machines are the crankcase, crankshaft, motor and condenser. Even the crankcase and condenser are designed so that the same tools can be used to produce all three sizes. The common parts feature is important from a service angle as the renewal parts, ordinarily carried by the field repair and maintenance organizations, are few and an ample working stock can be easily maintained.

***New Condensing Unit Compared With Previous Design***

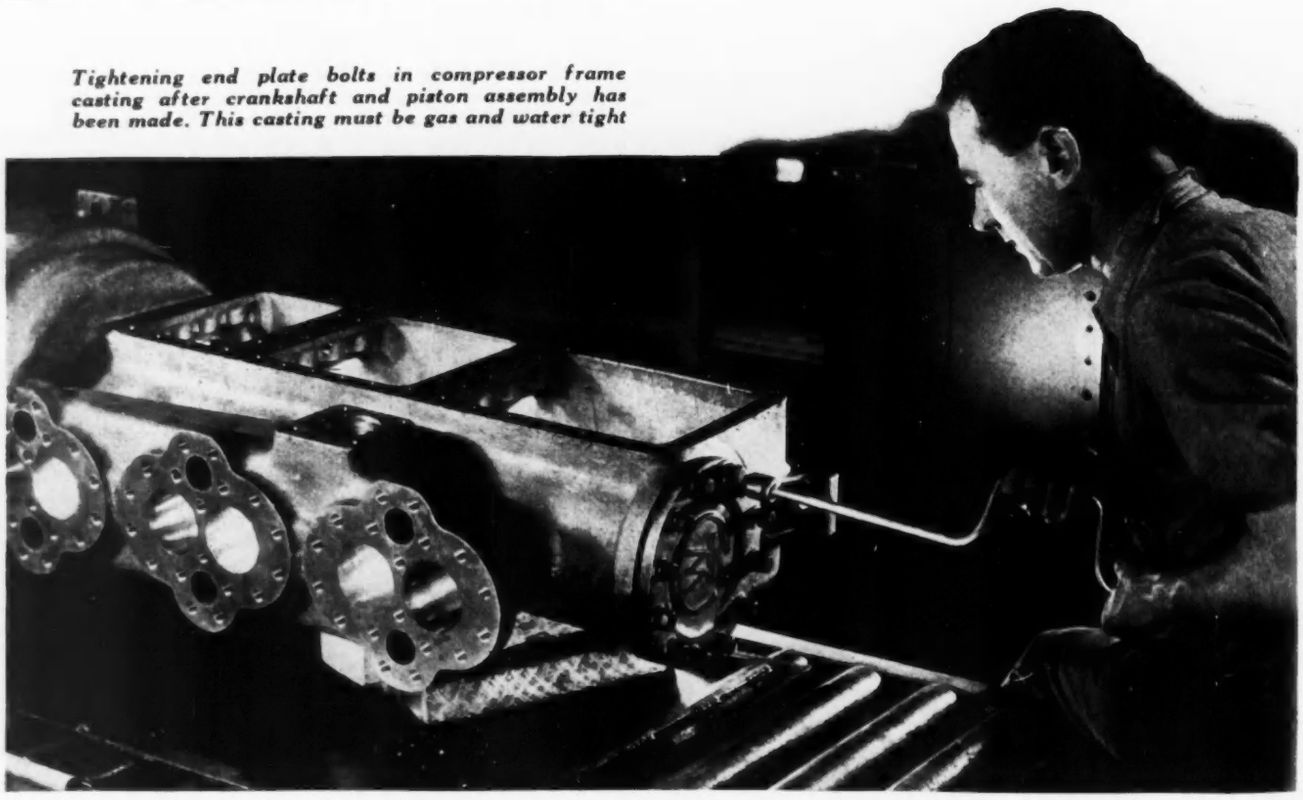
| | New Design 4-Cylinder Sealless Design | Former 6 Cylinder Design |
|---|---|---|
| Weight | 1,600 lb. | 3,400 lb. |
| Length | 54 in. | 76¾ |
| Width | 26 in. | 30⅞ |
| Height | 40 in. | 43¼ |
| R.p.m. | 1,160 | 870 |
| B.t.u./Hr. Capacity | 156,000 | 165,000 |
| Rated Motor Hp | 15 | 20 |
| Sq. In. Gasketed Surfaces | 50% | 100% |

## *Reduction in Weight and Size*

The accompanying table is a comparison for the 4-cylinder new design against the previous 6-cylinder design of comparable cooling capacity, it shows that: (1) The weight has been considerably reduced because the general construction is such that the need for the heavy, stiff bedplate is eliminated since it is not necessary to align the compressor motor. The wall thickness is the same as the old design. (2) The overall size has been reduced somewhat because of the method of mounting the motor, compactness in design, and absence of bedplate. (3) The r.p.m. is higher but not higher than in some previous designs. (4) The capacity is about the same with a 25 per cent saving in rated motor hp., and a 50 per cent reduction in the area of gasketed surfaces.

After assembly the compressor complete with condensor is completely tested with its control for performance. The rating of the larger units is obtained by loading with a steam calorimeter. Dry superheated steam is admitted to the calorimeter and the condensate measured. From the data noted the rating is easily obtained. The smaller units are loaded by means of an electric calorimeter, the electrical energy being converted into heat instead of using steam as is done with a steam calorimeter. Before leaving the test floor the units are filled with sufficient refrigerant to give a gas pressure slightly greater than atmospheric, using the refrigerant for which the unit is designed. It is then painted, boxed and ready for shipment.

***Tightening end plate bolts in compressor frame casting after crankshaft and piston assembly has been made. This casting must be gas and water tight***



# How to Make Alignment Charts

M. G. VAN VOORHIS

## Graphical and plotting methods for developing the curved scales in nomograms by use of either right angle or oblique coordinates, and also charts of quadratic equations

### PART VI

IT HAS been shown in some of the preceding nomograms that curves may be plotted on axes which are not at right angles to each other as well as on squared paper and that the laws of analytic geometry hold in a similar manner. Since it is not relevant to the subject to go into the theory, it will be assumed that it is accepted. The use which is made of this fact is to raise the $w$ scale as many units $f(v)$ or $f(w)$ units as desired, assuming that the $x$ axis still passes through the zeros of the two function scales. However, it is still practical to plot the curve on rectangular coordinates.

In Fig. 38 no change has been made in the direction of the $y$ axis; therefore, all values of $x$ are plotted the same as in Fig. 37, page 270 in the July issue. An increment must be added to $y$ values dependent upon the number of units, $s$, that zero of the $w$ scale is raised. The new ordinate, $y'$, is determined by the formula $y' = y - sx$. Using the same formula as in Fig. 37 and assuming the $L$-scale is raised ten units, the curve would be plotted from the following table which was derived from Fig. 37.

| $V$ | $x$ | $y'$ | $V$ | $x$ | $y'$ |
|---|---|---|---|---|---|
| 1..... | —0.0139 | 0.166 | 10..... | —0.488 | 6.28 |
| 3..... | —0.0953 | 1.176 | 15..... | —0.667 | 8.72 |
| 5..... | —0.211 | 2.65 | 20..... | —0.772 | 10.22 |
| 7..... | —0.332 | 4.22 | | | |

The completed nomogram is shown in Fig. 38. While this form is obviously the most satisfactory, it should not be assumed that this is true in every instance.

The ranges of $f(v)$ and $f(w)$ may be widely different making it desirable to have scales of the same length but with different size units. In this instance, the type equation is written, $Rf_1(u)\ f_3(v)/R + f_2(u)\ f_4(w) = 1$ when the $f_3(v)$ scale has $R$ times the range of $f_4(w)$ scale.

Therefore:

$$x = \frac{-f_2(u)}{f_5(u) + f_2(u)} \qquad y = \frac{1}{f_5(u) + f_3(u)}$$

where $f_5(u) = Rf_1(u)$ and $f_3(v)/R = f_6(v)$. The size of the unit of $y$ is the same as a unit of $f_6(v)$ and $f_4(w)$ values.

In an example, when the range of $H$ is from 0 to 100 and that of $L$ from 0 to 10, the $H$ scale has 10 times the range of the $L$ scale; therefore $R = 10$. The equation would be written:

$$\left(\frac{10}{0.0274V^2}\right)\frac{H}{10} + \left(\frac{0.0141V^{1.83}}{0.0274V^2}\right)(-L) = 1$$

The variation will give the following equations for $x$ and $y$

$$x = \frac{-0.0141V^{1.83}}{10 + 0.0141V^{1.83}} \qquad y = \frac{0.0274V^2}{10 + 0.0141V^{1.83}}$$

$$f_6(v) = H/10 \qquad f_4(w) = -L$$

Both $f_6(v)$ and $f_4(w)$ still vary from 0 to 10, therefore the $f_4(w)$ scale is raised 10 units as before.

In Fig. 39 are shown points plotted from data given in the following table which are computed from the above equations.

| $V$ | $y$ | $x$ | $y'$ |
|---|---|---|---|
| 10..... | 0.250 | —0.0871 | 1.121 |
| 20..... | 0.819 | —0.253 | 3.35 |
| 30..... | 1.444 | —0.414 | 5.58 |
| 40..... | 1.992 | —0.555 | 7.54 |
| 50..... | 2.432 | —0.645 | 8.88 |
| 60..... | 2.79 | —0.715 | 9.94 |

Assuming that the range of $H$ is from 0 to 10 and that of $L$ from 0 to 100, as in Fig. 40, the equation can be written:

$$\left(\frac{1}{0.274V^2}\right)(10H) + \left(\frac{0.0141V^{1.83}}{0.0274V^2}\right)(-L) = 1$$

The equations for $x$ and $y$ would then be

$$x = \frac{-0.141V^{1.83}}{1 + 0.141V^{1.83}} \qquad y = \frac{0.274V^2}{1 + 0.141V^{1.83}}$$

$$f_6(v) = 10H \quad \text{and} \qquad f_4(w) = -L$$

In this instance the $y$ axis is divided into 100 units ($f_6(v)$ varies from 0 to 100) for the function scales and the $L$ scale is raised 100 units to be opposite the $H$ scale ($s = 100$).

In Fig. 40 are shown points plotted from data given in the following table which were computed from the above equations.

| $V$ | $y$ | $x$ | $y'$ |
|---|---|---|---|
| 1..... | 0.240 | —0.124 | 12.60 |
| 2..... | 0.730 | —0.333 | 34.08 |
| 3..... | 1.199 | —0.513 | 52.5 |
| 4..... | 1.578 | —0.641 | 65.7 |
| 5..... | 1.861 | —0.728 | 74.7 |
| 6..... | 2.08 | —0.790 | 81.1 |
| 8..... | 2.39 | —0.864 | 88.8 |
| 10..... | 2.60 | —0.905 | 93.1 |
| 20..... | 3.15 | —0.972 | 100.3 |

***Nonograms for*** $\frac{f_1(u)}{f_3(v)} + \frac{f_2(u)}{f_4(w)} = 1$

This form is essentially the same as a previous type $f_1(u)\ f_3(v) + f_2(u)\ f_4(w) = 1$ but considered with reciprocal scales of the functions of $v$ and $w$. It is when the scales of $v$ and $w$ are both reciprocal that a

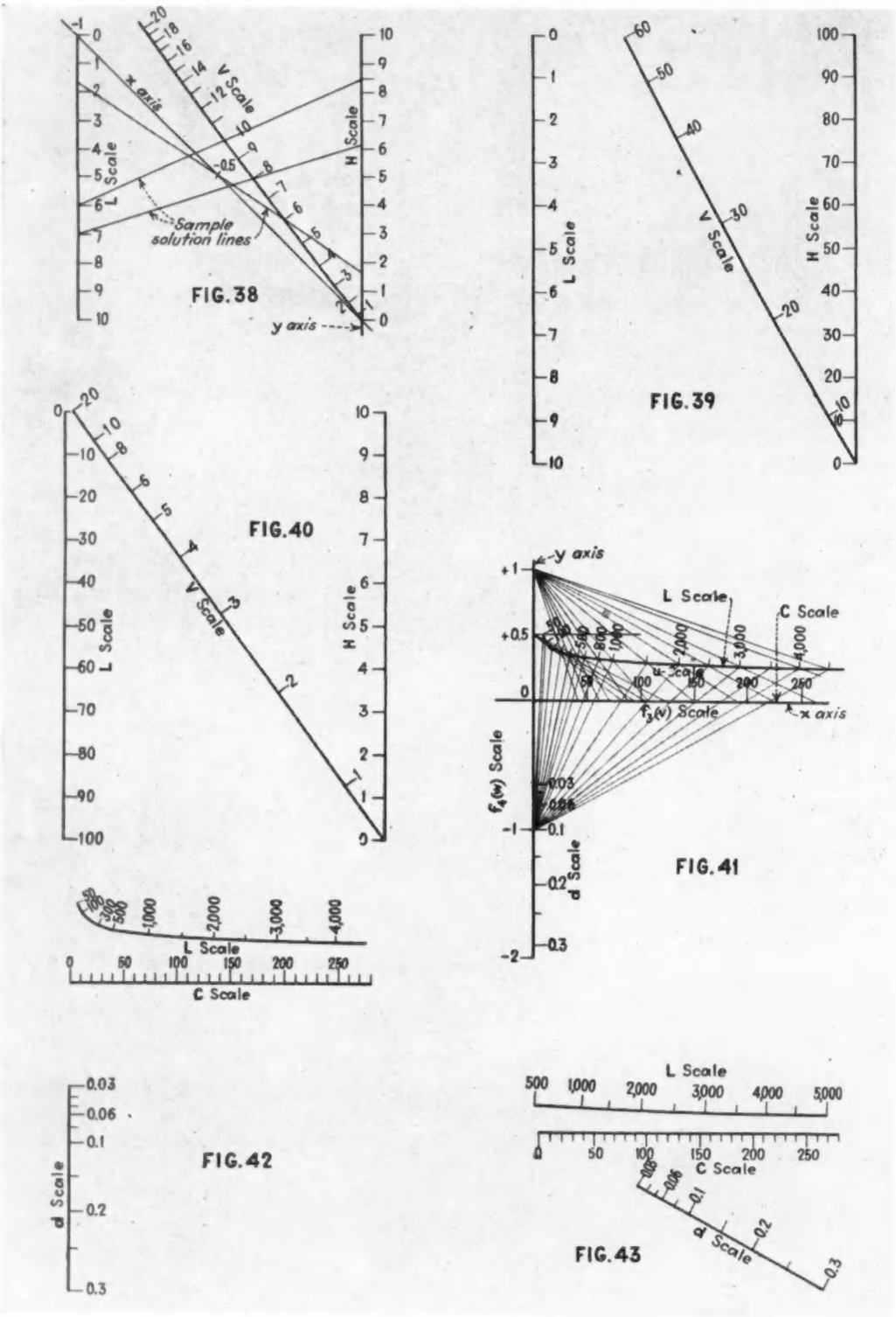



*FIG. 38*
***The curved V scale is plotted from calculated values derived from the table for Fig. 37 nomogram***

*FIG. 39*
***The range of the H scale is from 0 to 100 while the L scale has a 0 to 10 range***

*FIG. 40*
***The L and H values are reversed from Fig. 39, the H scale, having a range of 0 to 10 and a 0 to 100 range for L scale***

*FIG. 41*
***Graphical method for laying out nomogram for electrical capacity of a vertical wire relatively high above ground***

*FIG. 42*
***The u scale values are plotted on the coordinates formed by the $f_3(v)$ and $f_4(w)$ scale lines***

*FIG. 43*
***Oblique coordinates can be used for locating the u scale either graphically or by plotting***

new method is to be sought. It occasionally happens that one is reciprocal and the other is not. In this event a reciprocal scale cannot be avoided but it is possible to choose which variable would be least objectionable on a reciprocal scale.

The graphical method of laying out such a monogram is shown in Fig. 41 in which two straight lines intersecting at right angles are drawn. One line is uniformly subdivided with a scale of $f_3(v)$ and the other with a scale of $f_4(w)$ so that zero of both scales is at the intersection of the two lines. The size of each scale is independent of the other and may be as large as the space allows for the given limits.

To locate the $u$ scale line, a value of $f_3(v)$ or $f_4(w)$ is selected and the original equation is solved for $f_4(w)$ or $f_3(v)$, respectively, for a set of values of $u$ which are to be placed on the $u$ scale. This process is repeated for another value of $f_3(v)$ or $f_4(w)$ using the same values of $u$. The intersections of lines corresponding to the same value of $u$ will locate values of $u$ on the $u$ scale. The function scales are then replaced with scales of the variable.

As an example, the electrical capacity of a vertical wire relatively high above ground is determined by the formula, $C = 0.2416\ L/\log_{10}(2L/D)$, in which $L$ = length in cm.; $d$ = wire diameter in cm.; and $C$ = capacity in microfarads. It may be written:

$$\frac{0.2416L}{\log 2L/C} + \frac{1/\log 2L}{1/\log d} = 1$$

Therefore $f_3(v) = C$ and $f_4(w) = 1/\log d$.

A set of values of $C$ can be determined for $f_4(w) = 1$ and for $f_4(w) = -1$. These are tabulated below. For low values of $L$, $C$ is determined for $f_4(w) = 0.5$.

| $L$ | $C_{0.5}$ | $C_1$ | $C_{-1}$ |
|---|---|---|---|
| 50 | $\infty$ | —— | 6.04 |
| 100 | 80.3 | —— | 7.32 |
| 200 | 80.3 | —— | 13.41 |
| 300 | 93.2 | —— | 19.17 |
| 400 | 101.4 | —— | 27.75 |
| 500 | 120.8 | —— | 30.2 |
| 600 | —— | 69.7 | 35.5 |
| 800 | —— | 87.7 | 46.0 |
| 1,000 | —— | 105.0 | 56.1 |
| 1,500 | —— | 146.3 | 81.0 |
| 2,000 | —— | 185.6 | 105.0 |
| 2,500 | —— | 223.9 | 128.5 |
| 3,000 | —— | 261 | 151.6 |
| 3,500 | —— | 297 | 174.5 |
| 4,000 | —— | 333 | 197.0 |
| 4,500 | —— | 368 | 219.5 |

The plotting method in Fig. 42 is as simple as the graphical method. The $f_3(v)$ scale is uniformly divided on the $x$ axis with zero at the origin and the $f_4(w)$ scale is uniformly divided on the $y$ axis with zero at the origin. The size of each scale is independent of the other and may be as large as the space allows for the given limits.

For the location and subdivision of the $u$ scale let $x = f_1(u)$ and $y = f_2(u)$. A table is made of the values of $x$ and $y$ for each value of $u$ which it is desired to show on the $u$ scale line. The values are plotted on the coordinates formed by the $f_3(v)$ and $f_4(w)$ scale lines using the scales selected for each. The function scales of $v$ and $w$ are then replaced with scales of the variable.

The foregoing example under Fig. 41 is ideally suited to this type of equation since the reciprocal logarithmic scale works out well. Let $x = 0.2416\ L/\log 2L$ and $y = 1/\log 2L$.

| $L$ | $x$ | $y$ | $x'$ (Fig. 43) |
|---|---|---|---|
| 50 | 6.04 | 0.5 | —63.06 |
| 100 | 10.5 | 0.434 | —50.25 |
| 300 | 26.1 | 0.360 | —24.3 |
| 500 | 40.3 | 0.333 | — 6.3 |
| 1,000 | 73.2 | 0.303 | +30.8 |
| 1,500 | 104.2 | 0.288 | 63.9 |
| 2,000 | 134.1 | 0.277 | 5.3 |
| 2,500 | 163.3 | 0.270 | 125.5 |
| 3,000 | 191.8 | 0.265 | 154.7 |
| 3,500 | 220.0 | 0.260 | 183.6 |
| 4,000 | 247.6 | 0.256 | 211.8 |
| 4,500 | 275 | 0.253 | 239.6 |

The points are plotted in the nomogram in Fig. 42, from the data given in the above table which are computed from the above equations. The illustration shows how the nomogram should appear in completed form except for insufficient scale divisions. By using the formula for $x$ it is a simple matter to fill in scale divisions on the $L$ scale.

Oblique coordinates may be used, determined either graphically or by the plotting method. The relative positions of the $v$ and $w$ scales should be known before starting, since the $w$ scale may be on the $+y$ axis, in which instance the $y$ axis would swing in the other direction. In the example illustrated in Fig. 43 the $f_4(w)$ scale of Fig. 42 is swung so that $f_4(w) = 2$ is moved to the right 280 units. This means that $x$ values in the above table must be decreased by 140 $y$ to give the $x'$ values.

The theory of the construction of nomograms for equations containing one function of two variables and two functions of the third variable and, in fact, for any of the parallel line nomograms considered up to this point, is derived from the equation of a straight line in rectangular co-ordinates. One form of this equation is:

$$\frac{y - y_1}{x - x_1} = \frac{y_1 - y_2}{x_1 - x_2}$$

In Fig. 44 a point corresponding to $x,y$ is on the curve at the right. The point corresponding to $x_1$, $y_1$ is at $v$ and that corresponding to $x_2$, $y_2$ is at $w$. The equation then can be written:

$$\frac{y - v}{x - O} = \frac{v - w}{O + 1}$$

Solving for $y$,

$$y = v + vx - wx$$
$$\text{or } y = v(x + 1) - wx$$

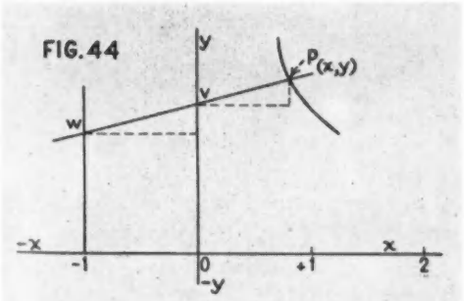



*FIG. 44*

***Numerous charts can be worked out from (y—u) / (x—O) = (v—w)/(O+1) for different nomographic positions of $f_3$ (v) and $f_4$ (w) scales when the equation is written in the form of $f_1(u)\ f_4w + f_2(u)\ f_4w = 1$***

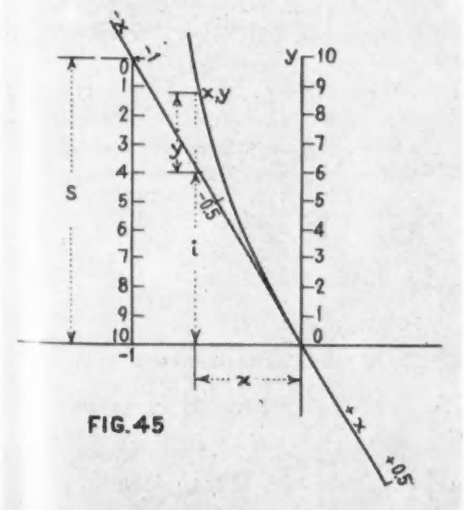
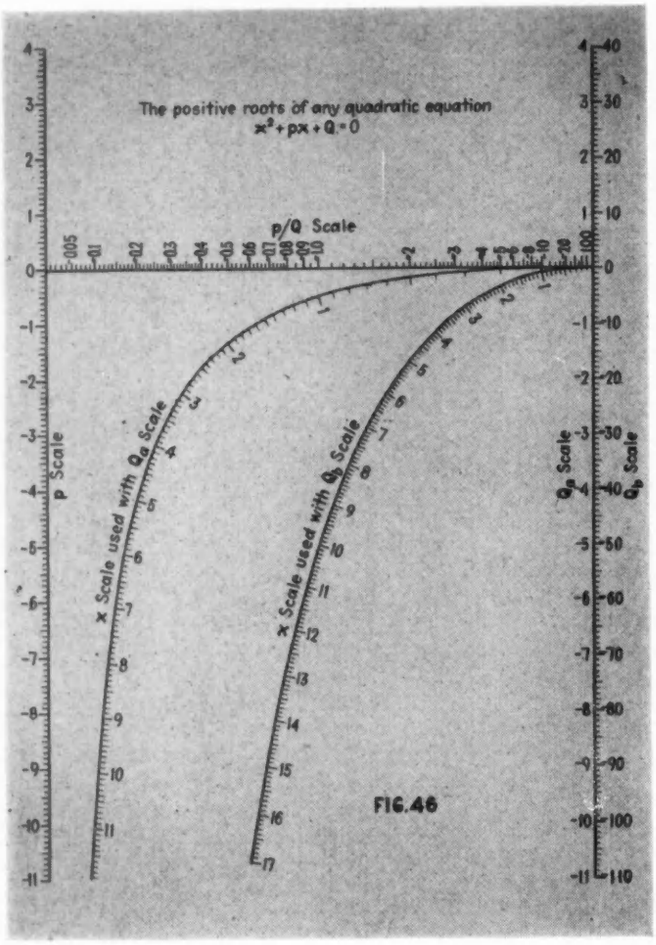



*FIG. 45*

***Showing the method of plotting oblique axes on rectangular coordinates whereby similar triangles are used for solution of the formula***




Dividing through by $y$; then

$$1 = \frac{x+1}{y}v - \frac{x}{y}w \text{ or } \frac{x+1}{y}v + \left(\frac{-x}{y}\right)w = 1$$

Since $v$ and $w$ represent values on the $y$ axis and the line $x = -1$, respectively, and the two functions of $x$, $y$ represent values on the curve, an equation which is written in the form, $f_1(u)\ f_3(v) + f_2(u)\ f_4(w) = 1$, can be solved by a straight line crossing scales of $f_3(v)$ and $f_4(w)$ on the $y$ axis and the line $x = -1$, respectively, and a curve plotted from values of $x$ and $y$ determined from the relations:

$$(x+1)/y = f_1(u) \text{ and } -(x/y) = f_2(u)$$
$$(1/y) + (x/y) = f_1(u)$$
$$1/y = f_1(u) + f_2(u) \qquad y = 1/[f_1(u) + f_2(u)]$$
$$x = -yf_2(u) = -f_2(u)/[f_1(u) + f_2(u)]$$

Similar equations can be worked out for any other positions of the scales of $f_3(v)$ and $f_4(w)$ such as the $y$ axis and $x = +1$, or $x = -1$ and $x = +1$. It is sometimes possible with these alternatives to simplify the computations.

A check on the location of the $u$ scale is afforded by an inspection of the equation for $x$. With signs the same as above it is obvious that no value of $x$ can be greater than unity and all values lie between 0 and —1. Therefore it is possible to visualize the positions when the plotting method is used, before drawing any line.

Plotting in oblique axes on rectangular coordinates is demonstrated in Fig. 45. By similar triangles it is obvious that $s/l = i/x$. Therefore $i = sx$ and $y' = y + sx = y - s\ (-x)$.

Many interesting examples of nomograms of this type have been constructed. That for the quadratic equation is well known. Its usual form is that shown in Fig. 46. A variation which is of interest brings out another method of plotting the $u$ scale.

***FIG. 46***
***A nomogram of the quadratic equation $cu^2 + u + d = 0$ laid out from the rewritten equation $d(-1/u) + c(-u) = 1$***

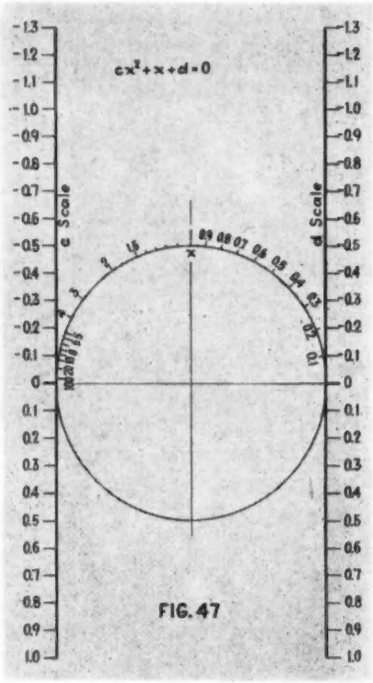



***FIG. 47***
***Location of the points on the arc of the equation for a circle when $x = -u^2/(u^2 + 1)$ and $y = -u/(u^2 + 1)$. Positive values of x are shown on the circle***

The equation, $cu^2 + u + d = 0$, may be written $d(-1/u) + c\ (-u) = 1$ in which $(x+1)/y = -1/u$ and $-x/y = -u$. Instead of solving for $x$ and $y$, $u$ may be eliminated in combining the two equations and there is left an equation for the line on which the $u$ scale is placed. Obviously, $(x+1)/y = -y/x$. Clearing fractions the equation becomes $-y^2 = x^2 + x + (\frac{1}{2})^2 - (\frac{1}{2})^2 = (x + \frac{1}{2})^2 - (\frac{1}{2})^2$. When the equation is written: $(x + \frac{1}{2})^2 + y^2 = (\frac{1}{2})^2$, it is recognized as the equation of a circle with its center at $x = -\frac{1}{2}$ and $y = 0$. The positive values of $x$ are shown in Fig. 47. By using the equations: $x = -u^2/(u^2 + 1)$ and $y = -u/(u^2 + 1)$ points on the arc may be located.

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**What German machine tool designers have developed in hydraulic drives is always a subject of great interest. September *Product Engineering* will have a feature article comparing the efficiencies of hydraulic spindle drives with gear drives, written by Dr. Max Kronenberg, one of the outstanding machine tool designers of Germany.**

# QUESTION AND COMMENT

## *Relation of Industrial Designer to Established Engineering Departments*

*To the Editor:*

● Editorially you have given much consideration to problems of industrial design and the function of the industrial designer. The writer would like a reliable summation of the duties of an industrial designer, more particularly his scope of authority and relations to established engineering departments dealing with mechanical design.

—GEO. B. WRIGHT
*South Norwalk, Conn.*

*To the Editor:*

● In modern manufacturing the industrial designer functions in close cooperation with the established engineering department and not independently. In fact, the closer the cooperation, the better the results. The industrial designer whose field embraces a latitude from lipsticks to steam shovels cannot possibly be expected, and is not expected, to have an intimate knowledge of all manufacturing problems involved in the production of a specific item. However, an engineering training enables him to grasp quickly the technical limitations so as to cooperate with the engineering department. They both speak the same language. His task is to develop the best possible design within the limits of practicality, and the engineering department will supply him with all information pertaining to existing manufacturing facilities and with the economics of the problem under consideration.

The modern engineering department will be quick to realize that the industrial designer comes into the picture as a friend with no intention to upset anything. He is not trying:

1. To supersede the engineering department. Even if he were able to do so, he would have no such interest.
2. To take credit from the engineering department for a successful design that was developed in collaboration.
3. To hint to the management that the department has obsolete design conceptions.

On the contrary, he is trying:

1. To help the engineering department become more actively style-minded, to inject new life and activity, and to discard possible design inhibitions.
2. To suggest materials and practices used successfully in totally unrelated industries, of which the engineering department may not be aware.
3. To prove to the management that the industrial designer has a salutary influence on the engineering department without in any way upsetting or modifying its personnel or its layout.

This last item, most vital to the industrial designer's success, means a satisfied management; a renewal of the contract; an engineering department whose prestige has increased; cordial relations with the appreciative engineers; and from then on smooth running, pleasant and profitable relation.

May I quote in this respect an excerpt from a letter written to me several years ago by one of the Sears, Roebuck & Company executives:

"Due to the manner in which you worked with us, I can conservatively state that it was the easiest job of correlation between designer and manufacturer that we have ever had."

Our business connection with this firm has greatly developed since that time and the sales figures have conclusively proven from year to year the soundness of the cooperation. At no time have I found that credit was diverted from the admirable work being done by the engineering departments because of an industrial designer entering upon the scene.

—RAYMOND LOEWY
*New York, N. Y.*

*To the Editor:*

● We do not attempt to design a product mechanically, being interested particularly in designing it from the appearance angle. It is necessary, of course, that we cooperate with the engineering department inasmuch as the design suggestions we make must be worked out mechanically. If in the process of working on appearance design we are able to make suggestions regarding mechanical improvements, we do so, but we do not consider this the main object of our work.

Before starting work, we go through the factory and discuss the problem thoroughly from the manufacturing as well as the merchandising angles. Although our staff consists of trained engineers and architects, all working drawings are made by the factory from the presentation we made of our designs—either rendered perspectives, plans and elevations or models.

The reason for having the manufacturer make working drawings, which are referred back to us, is that any "snags" which arise regarding production problems can be recognized immediately.

We have found that in working with manufacturers, there is no conflict between our work and that of the established engineering department in the company. In a sense we become a part of the company and while cooperating with them we do not supplant but supplement the engineering department. Our method of work is based on the close cooperation between the various departments of a company and ourselves, as that is the only way in which satisfactory results are obtained.

—HENRY DREYFUSS
*New York, N. Y.*

*To the Editor:*

● As far as the established engineering department is concerned and relation of the industrial designer to such a department, I believe there should be no conflict whatsoever. In our own experience we have dealt with dozens of firms who have had engineering facilities of a high order. We have rarely ever had trouble in gaining their confidence as soon as they knew how we worked. In fact the better the engineer, the more he welcomes assistance on purely visual problems. An engineer should not be expected to produce an artistic design, for his training probably is not along these lines.

Our method of working on any styling problem usually is divided into three stages. The first stage consists of rough sketches which are discussed freely with the executive, sales and engineering departments. Many ideas may be involved in these sketches which may not be embodied in the final design. However, we do not present sketches unless we know that they are possible solutions. This is difficult and important, as it presupposes having become familiar with the facilities of the plant and with the sales requirements. During this time the engineers are consulted continuously, and I believe because of sympathetic understanding of his problems we are able to overcome any antagonism he may feel, even though unexpressed.

The second stage of the work is usually the preparation of finished perspective drawings in color or the preparation of scale models. The build-

ing of a model is not attempted until we have worked out most of the practical details and know we are on the right track.

The final stage of the development is the presentation of finished dimensioned mechanical drawings on all factors which have to do with appearance. By this I do not mean that they are shop drawings, but they are a guide to the preparation of the final shop drawings. To make them of value, an experienced engineer is retained to supervise the production of these drawings, and in some instances they are actually used in building the experimental models. However, the determination of the gage of sheet metal, thickness of castings and other factors are the ultimate responsibility of the company's own engineers.

In this way, by close cooperation in all three stages of developing a product, we collaborate fully with the resident engineers, and yet leave none of the factors affecting appearance to guess work. —HAROLD VAN DOREN
*Toledo, Ohio*

## *Calculating Belts for Stepped Pulleys*

*To the Editor:*

● The article "Belt Lengths and Pulley Diameters" for finding the diameters of cone pulley steps, on pages 199 and 200 of the May issue is interesting. However, the method is incorrect and misleading. The belt length is calculated without taking into consideration the belt thickness. It is necessary to add one thickness of belt to the diameters of each step on the cone pulleys before making calculations, otherwise the belt will be too short. As on those pages the length of belt as found was 140.375 in. The actual length necessary using a belt 0.2 in. thick for 2 to 1 ratio is 141.044 in.; for 3 to 1 ratio is 141.029 in.; for 4 to 1 ratio is 141.037 in., and for 5 to 1 ratio is 141.121 in. The belt therefore would be more than $\frac{5}{8}$ in. too short and if it could be placed on the pulleys at all, it would be too tight. This places a heavier load on the bearings and certainly does not prolong the life of the belt or bearings.

To illustrate the procedure of calculation when belt thickness is added, for the small pulley the diameter will be 8.7 in. The circumference will be 8.7 in. x 3.1416 equal 27.332 in. Therefore, as the angle of the arc of contact for this pulley 170 deg. and 14 min. as in the sketch, length of belt on the small pulley is $(27.332 \times 10{,}214)/21{,}600 = 12{,}924$ in., where 10,214 equals the number of minutes contained in 170 deg. and 14 min. Calculating the large pulley in a similar manner, the total length will be:

$$99.637 + 28.483 + 12.924 = 141.044 \text{ in.}$$

The other three sets of steps are worked out the same way.

EWARD J. RANTSCH
*Hollis, N. Y.*

## *Design of Solenoid*

*To the Editor:*

● The solenoid about which J. M. Marateck inquires in the May issue will have considerable more load than the spring load alone. Depending on its operating position, the plunger weight, or the pull against friction must be considered. To eliminate chattering when working on a.c. the pull of the coil should exceed the sum of the loads against which it works. The design should be strong enough but not wasteful. Since the coil works against a spring, a greater pull will be required at the end than at the beginning of the stroke, making it advisable to insert a fixed steel plug $\frac{1}{2}$ in. long flush with the end of the coil. The plunger butts against the plug at the end of its stroke.

Basing the design on the features enumerated and assuming that a $\frac{1}{2}$-in. diam. core rod of mild steel, is used, a coil made of about 1,800 turns of number 25 enameled copper wire should give enough pull, and will approximately fill the $1\frac{3}{8}$ in. diameter by $2\frac{1}{8}$ in space allowed, leaving room for a fiber winding spool. This coil will have a resistance of approximately 15 ohms and an inductance of about 0.12 henry when the plunger is against the core stop, the inductance, of course, varying with the position of the plunger.

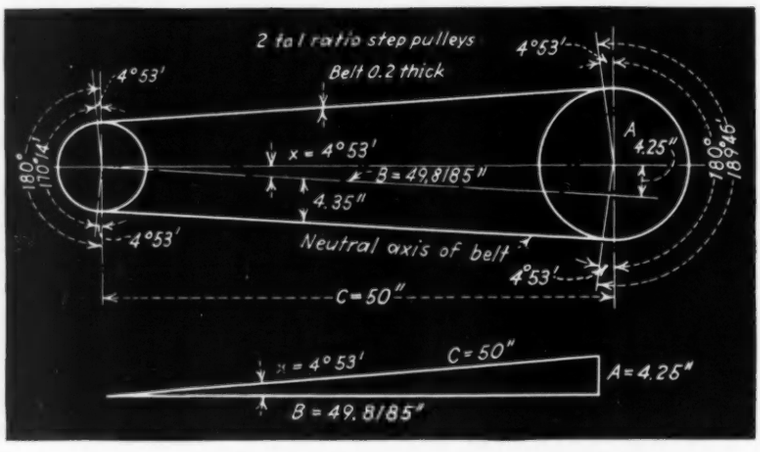
When used with d.c. current, an external resistance of about 35 ohms should be used. Whether on d.c. or a.c. the coil will heat up at the rate of about $1\frac{1}{2}$ deg. per sec. but this is not dangerous because it is to be used for only a few seconds at a time.

The pull $F$ in lb. on the plunger at any position of its stroke is given by the formula:

$$F = Ina\frac{\text{In}}{7{,}000{,}000\ L}\ 2 + \frac{0.009}{b}$$

$I$ = current in amp.
$n$ = number of turns in the coil.
$a$ = area of the core in sq.in.
$L$ = clearance between the plunger rod and core plug in in.
$b$ = length of the coil in in.

The factor 0.009 depends on the ratio of core diameter to coil length, the material of the core, and the degree of magnetic saturation of the core. —LEO G. HALL
*Allegan, Mich.*




***In calculating the length half of the belt thickness must be added to the radius of each pulley***

## *Synthetic Rubber Mixed With Cork*

*To the Editor:*

● For similar reasons as those mentioned in the article concerning rubber-cork combinations appearing in the May issue of *Product Engineering*, synthetic rubber is combined with cork. It has the advantage of resisting oil, gasoline and other influences destructive to natural rubber. The cork-rubber combination is more flexible and has greater tensile strength than cork alone. When a resilient material like rubber is subjected to pressure along one axis, the dimensions along all perpendicular axes are increased, the total volume remaining substantially constant. When certain percentages of cork are added the all-dimensional flow is lessened, making the material compressible without causing distortion in other than the direction in which the pressure is applied.

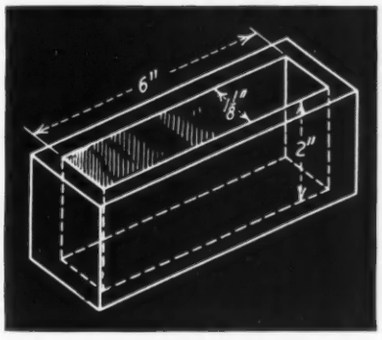
The sanded surface of such a vulcanized composition has a high coeffi-

cient of friction, which makes the material specially suitable for feed and friction rolls. Other applications include valve disks for oils and fuels, valve packing, gaskets for transformers and carburetors and nonskid wing walks for airplanes. It is interesting to note that gaskets of this material do not crumble when they are removed, and thus can be used over again. —E. H. CAUSEY
*E. I. duPont de Nemours & Company*

## What Dimension of Wall Thickness?

*To the Editor:*

● What would be the minimum wall thickness of a rectangular tube as illustrated, with one end closed and to withstand a pressure of 6,000 lb. per sq.in.? The inside dimensions are 6 in. long by $1\frac{1}{8}$ in. wide and 2 in. deep. The tube is to be made of machine steel. —H. E. BENTON
*Philadelphia, Pa.*




***What thickness of wall is required for this rectangular tube with closed end, to withstand 6,000 lb. per sq.in. pressure?***

## Screw Back-Lash Adjustment

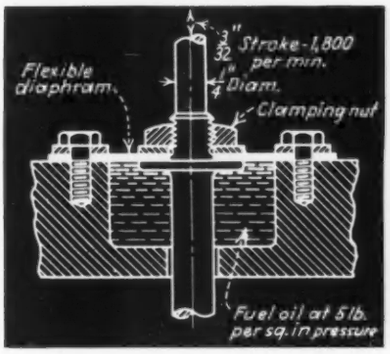
*To the Editor:*

● Can you give me a design of a bronze nut for a lead or feed screw in which the slack or wear can be taken up? Is it possible to incorporate this feature in a single piece that will give reasonably long wear? Could such a nut be made by soldering or sweating two pieces together and then boring, chasing and finish-tapping the hole center line at their joint and afterwards melting the solder, anchoring one half and adjusting the other half? I would also like to have some simple and efficient adjusting device for a lead or feed nut. —HENRY MARTIN
*Kensington, Conn.*

## 100 Per Cent Seal Required

*To the Editor:*

● A seal that is 100 per cent effective is required for a valve stem reciprocating at 1,800 strokes per min. having




***A method of sealing for 100 per cent effectiveness which failed after several weeks of operation at 1,800 strokes per min.***

$\frac{3}{32}$ in. length of stroke. It can be metallic or non-metallic but the seal should have a minimum life of at least one year. The sketch shows a method that failed. Can your readers suggest designs or materials? —S. P. LEE
*Sydney, N. Y.*

## Bearings Wanted

*To the Editor:*

● Can any reader of *Product Engineering* inform me if there is a self-lubricating non-metallic bearing made? If so, what are the physical and chemical characteristics, what is the maximum surface speed the material will withstand and on what equipment have such bearings been used?
—A. M. HOLLAND
*New York, N. Y.*

## Are Elliptical Gears In Modern Use?

*To the Editor:*

● I am wondering if elliptical gears are being used in any machines at the present time. Would such machines be "old timers" or does newly designed equipment contain this type of gear?
—A. H. SHUM
*Rockford, Ill.*

[Editor's Note: *Such gears are now in use and a list of some of their typical applications would undoubtedly be of general interest. We will be glad to hear from readers with reference to such applications.*]

## Chrome Plating Beryllium—Copper Molds

*To the Editor:*

● In the May issue of *Product Engineering* you published a letter written by E. N. Schmitt in which he suggests that chromium plating may be a means of prolonging the life of beryllium-copper molds. We are glad to advise you that we have several plated molds which are now in operation.

This plating was done purely as an experiment in connection with our general development in the production of cast beryllium-copper molds for use with plastics. The object of the experiment was to determine how well the molds would take the plating and how well the plating would stand up under continuous operation using various forms of plastics. There are certain plastics in use which, under some conditions, develop sticking troubles, and others which, because they contain sulphur or similar ingredients, cause staining or pitting of steel molds. Therefore, we felt that it would be desirable to have some knowledge of what could be done with chromium plating in the event it might become desirable to use cast molds with plastics of this type. It was really with that in mind that we undertook the experiment rather than any thought of prolonging the mold life.

So far the chromium-plated molds are working quite satisfactorily. They have not as yet been operated long enough for us to draw any conclusions as to their ultimate life. They have been put through several hundred cycles only but to date there is no evidence of any tendency for the plating to become scratched, peel or wear out.

Aside from the question of plating, it may be of interest that beryllium-copper molds in operation for several thousand cycles, with urea and phenolic types of molding compounds, show no evidence of surface abrasion or other failure.

We tried one experiment in which a beryllium-copper cavity and a hardened steel upper die were used. After 4,000 cycles the land of the steel upper die had failed to a marked degree, while the beryllium copper was still in perfect condition. It is assumed that this is because the beryllium copper has a compressive strength of 200,000 lb. per sq.in. throughout, which is in excess of that for the inner portion of a block of steel to which hardening does not penetrate.

—K. N. ATWATER
*The Gorham Company*

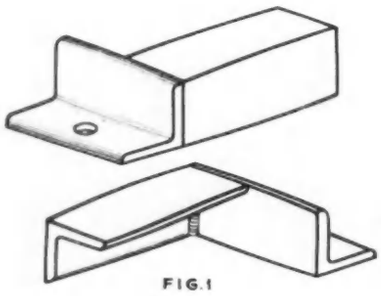


FIG.1

# *Typical Joints of Arc-Welded Structurals—I*

R. A. GAST
*The Lincoln Electric Company*

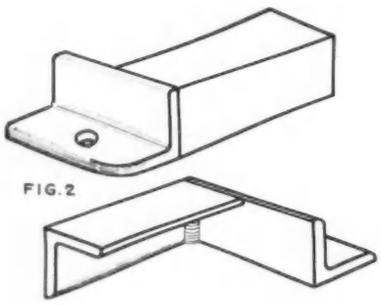


FIG.2

Examples of welded joints showing various methods of joining at right angles, standard structural angles, channels, H-sections, I-beams and combinations of these shapes. Additional examples will appear in September.

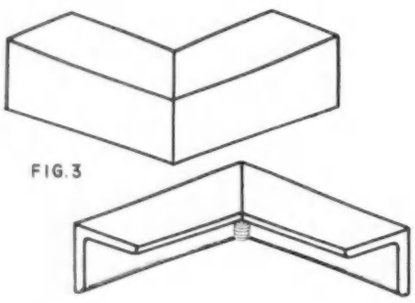


FIG.3

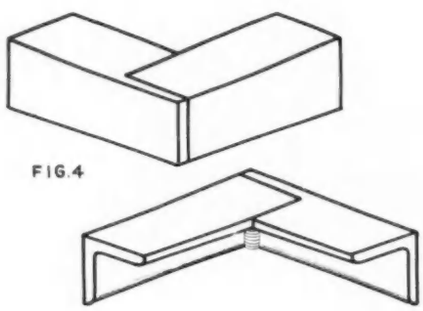


FIG.4

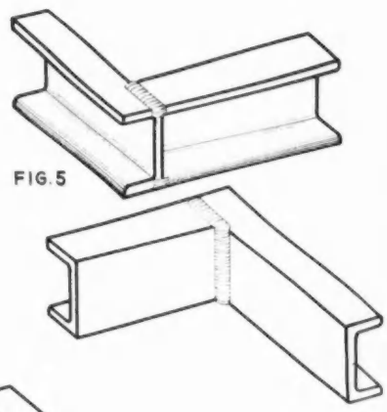


FIG.5

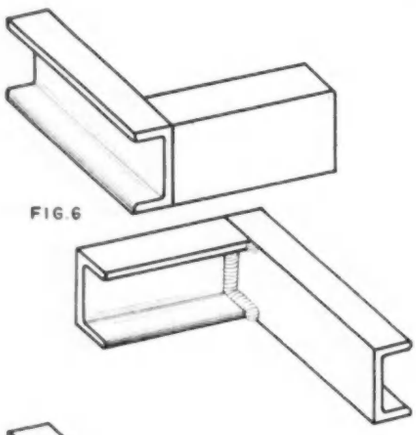


FIG.6

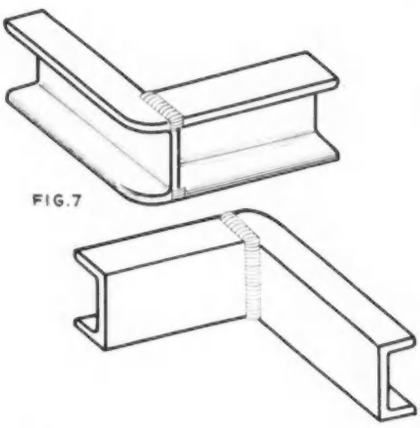


FIG.7

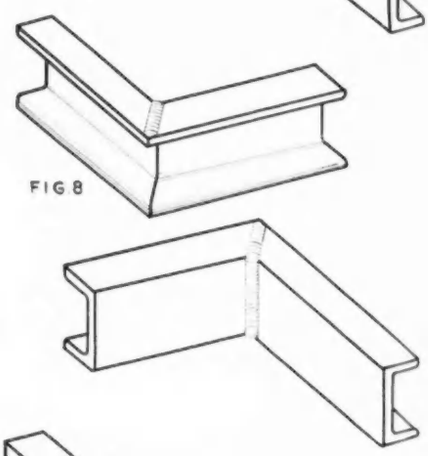


FIG.8

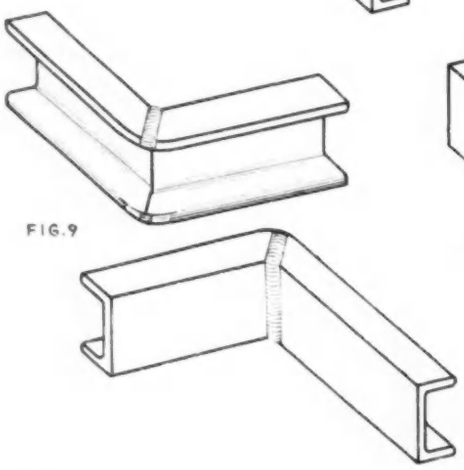


FIG.9

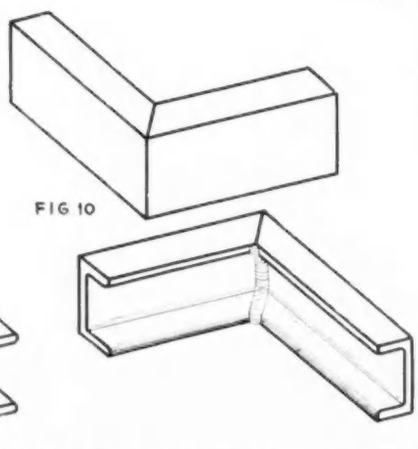


FIG 10

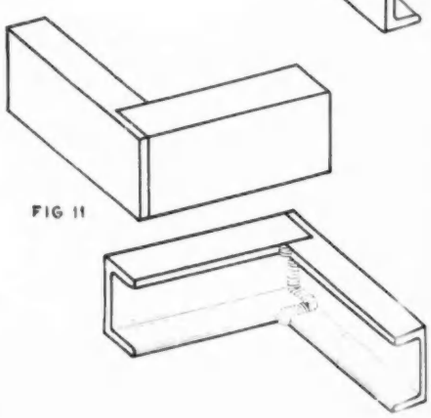


FIG 11

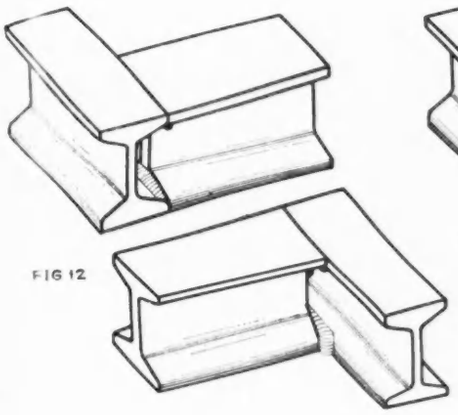


FIG. 12

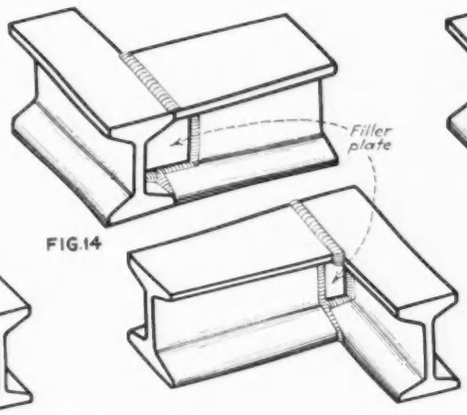



FIG. 14

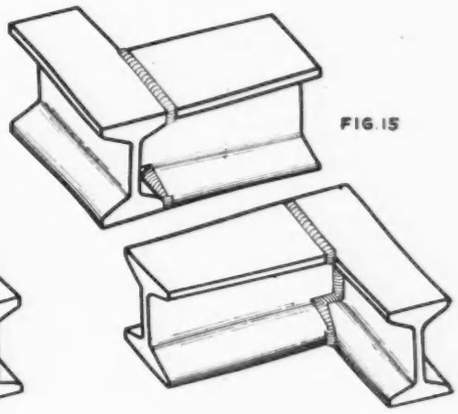


FIG. 15

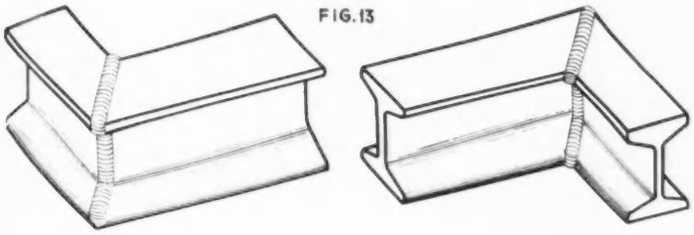


FIG. 13

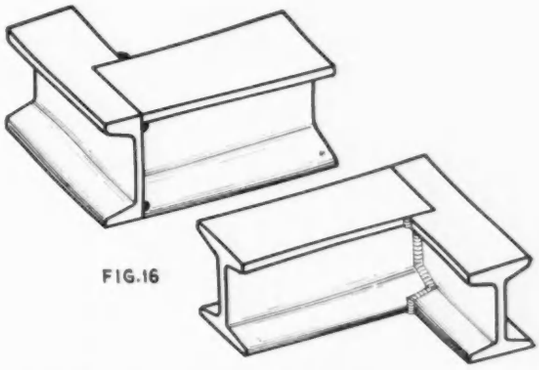


FIG. 16

The cost of preparing the pieces is somewhat less when they are cut at right angles than when mitered or set-in. However, in the two latter methods, the welds are longer and consequently stronger. Horizontal welding is less costly than vertical or overhead welding.

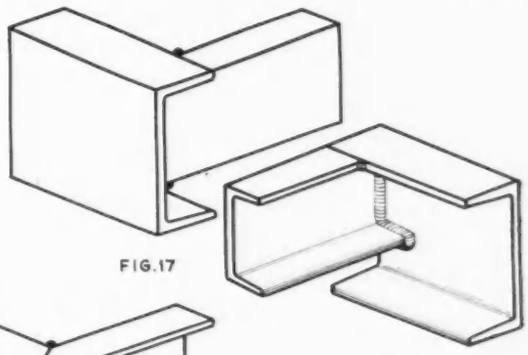


FIG. 17

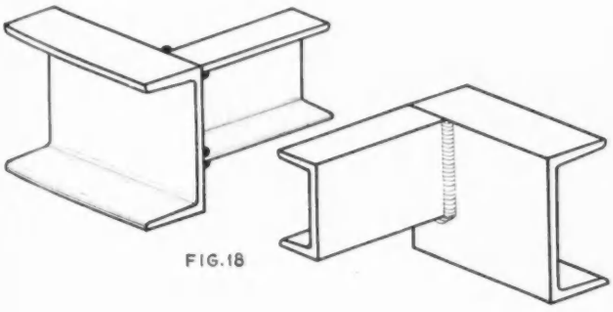


FIG. 18

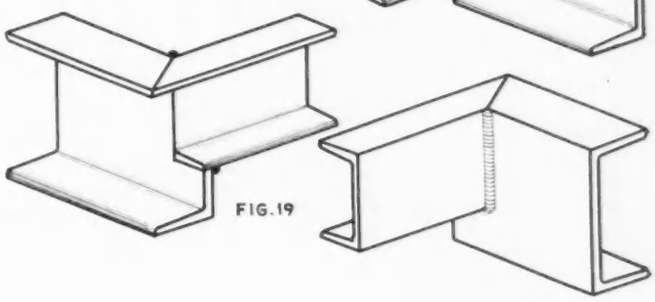


FIG. 19

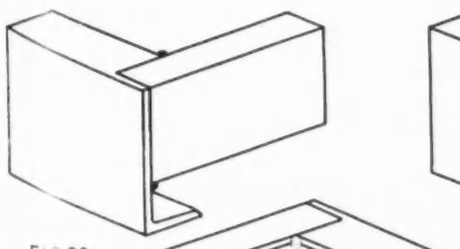


FIG. 20

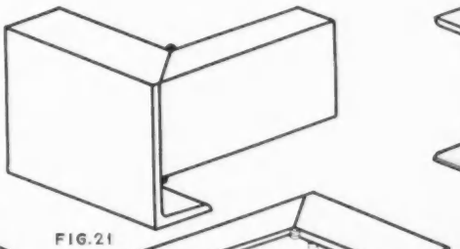


FIG. 21

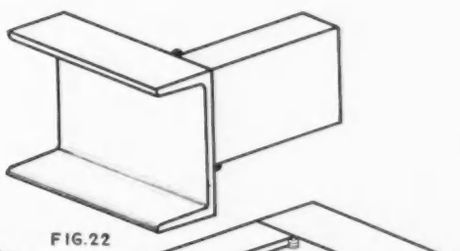


FIG. 22

# NEWS

## *Westinghouse Electrification Forum Attracts Machine Tool Designers*

FOR the purpose of giving machine tool designers a better working knowledge of electric motors and control devices, Westinghouse Electric and Manufacturing Company held a four-day (June 22–25) "Machine Tool Electrification Forum" at the East Pittsburgh works. Machine tool builders were invited to send, at their own expense, delegates to the conference. More than 50 engineers and executives representing almost as many manufacturers, attended. And judging by the numerous comments and opinions expressed by them, their time was well spent.

Technical sessions began Monday afternoon, June 22, with a series of short talks by Westinghouse engineers on the types and characteristics of a.c. and d.c. motors for machine tool applications. After each talk questions from the audience were answered by the speaker or other Westinghouse engineers. Some of the questions developed into brief discussions.

Typical of all the sessions, the papers delivered by the Westinghouse engineers were brief and to the point. R. S. Elberty, of the machinery electrification division of Westinghouse, handled the meetings and was commended for the manner in which he kept the sessions moving, not permitting a meeting to become dull by off-the-question or pointless discussions. It is also to the credit of Westinghouse that comparisons with competing equipment were carefully avoided at all times.

Sessions on Tuesday morning were devoted to discussions of control apparatus. In the afternoon the guests were taken on a trip through the East Pittsburgh works to see the motor and control manufacturing departments in operation.

Other sessions included the discussion of special control apparatus for machine tools; special problems involved in the application of motors to machine tools; a session on miscellaneous equipment such as gearmotors and including a paper on the possibilities of motor applications in machine tools. A special session was given over to discussing maintenance problems in relation to the selection of motors for machine drives. The closing session on Thursday was devoted entirely to an open discussion of machine tool electrification problems, including criticisms by the guests of available electrical devices.

Although all of the sessions were strictly business, numerous opportunities for social diversions were provided. These included an evening dinner followed by a trip through the Westinghouse Research Laboratories and a dinner-entertainment at one of the Pittsburgh country clubs. On each of the days luncheon was served to the guests in the executives' dining room, giving everyone an added opportunity for getting further acquainted and swapping ideas.

## *Welding Society Announces Program*

IN CONJUNCTION with the Metal Congress Exposition, to be held at Cleveland, Oct. 19–24, the American Welding Society will hold a series of meetings that will undoubtedly attract many designing engineers. In addition to 5 technical sessions, sponsored by the American Welding Society, there will be two joint sessions with the American Society of Mechanical Engineers and the latter society will be sole sponsor for two additional sessions.

Almost every type of welding and welded design will be covered in these meetings. High-speed motion pictures

***Machine tool designers from most of the progressive companies attended Westinghouse's four-day forum on motors and controls***

of various welding processes will be shown. Two sessions will be devoted to reports of the results of fundamental research in welding, in which will be included non-destructive testing of welds, impact tests of welds at low temperature, X-ray methods for studying stress relief and tests on welded structural brackets.

Papers to be presented at other sessions and of special interest to designing engineers will deal with the welding of copper and its alloys, resistance welding of dissimilar metals, and thermit welding.

At the joint sessions of the American Welding Society and the American Society of Mechanical Engineers, there will be three papers of special interest to designers. These will include analysis of stresses in welds, alloy steels and their weldability and welded heavy machinery.

The final program for this meeting will be published in the September number.

## *New Auto Plates To Be Anodized*

ALUMINUM license plates for automobiles, designed to last from 5 to 15 years, will be assigned to Connecticut automobile owners in 1937. These new plates will be stamped out of 0.050-in. sheet aluminum and will be Alumilited by the anodic process to increase hardness and durability for weather resistance. With a natural aluminum color background, black letters will provide sharp contrast with good visibility.

The original plate will be issued in the regular manner and each year a small metal insert designating the year, with a different color for each year, will be fastened on back of the plate, projecting through an opening to the front. It is thought that the new system will save the State approximately $125,000 a year.

## *Sloane Discusses Lack of Vision*

WITH festivities sponsored by the local chamber of commerce, Schenectady celebrated, on June 12 and 13, to commemorate the establishment of the electrical manufacturing industry in that city, 50 years ago. On June 14, 1886, Thomas Alva Edison took title to two abandoned shops of the McQueen Locomotive Company as a new location for the Edison Machine Works which later developed into the present General Electric Company.

One of the outstanding events of the two-day celebration was a formal dinner on Friday evening, June 12, attended by more than 500 invited guests. Among the chief speakers were Owen D. Young, chairman of the board of directors of the G. E. company, Charles A. Edison, son of the famous inventor, and W. S. Barstow, one of the Edison's early coworkers and now president of the Thomas Edison Foundation. Dr. Dixon R. Fox, president of Union College, acted as toastmaster. Quoting from Mr. Sloane's address:

"For the past six years, we have been advised on every hand to be realistic. This, I suppose, is the reaction against the alleged mirages of the 20's. Youngsters without practical experience were the ones in earlier generations who dreamed dreams and who complained of their elders for want of vision. In this generation these same youngsters profess to be hard-boiled realists who look at affairs today so objectively and with such assurance, that they warn their elders against being misled by the dreams of yesterday.

"The heart of this realistic philosophy seems to be that what is—is, and what is not—can't be; that our problem is to divide what we have and be content; that youth can no longer look forward to the *progress* which the past has known, that the research worker, the inventor and the engineer are enemies rather than benefactors of society; that we must surrender our visions and our hopes for the drab treadmill of a living in which the talented and untalented alike, the industrious and the lazy, the courageous and the cowardly shall have equality in recognition and division.

"Broadly speaking, there had not been a time during the past 50 years when anything manufactured by the General Electric Company was not, to some extent at least, obsolete by the time that it was put in service. So rapidly came invention and improvement, fostered by research, that in this growing art it could never be said and cannot be said today that what is—is. It could only truthfully be said that what is—is not, in the sense that what is, is too obsolete ever to be built again. The statement is also equally false that what is not—cannot be. The whole history of the 50 years shows that what is not—can be, and will be. Such is the craving of the human mind to solve the unknown; such is the human desire to make visions come true; such is the all-conquering spirit of men of this nation that their children shall have what their fathers lacked and wanted. This philosophy of an alleged realism and pessimism is the cause of most of the political, social and economic troubles of this day."

# *MEETINGS*

**Dairy and Ice Cream Machinery Manufacturers Association**—Atlantic City, N. J., Oct. 12–17. Roberts Everett, vice-president, 232 Madison Ave., New York, N. Y.

**National Safety Council Inc.**—25th congress and exposition. Atlantic City, N. J., Oct. 5–9.

**Society of Automotive Engineers**—Aircraft Production Meeting, Los Angeles, Calif., Oct. 15-17. John A. C. Warner, secretary and general manager, 29 West 39th St., New York, N. Y.

## *EXHIBITIONS*

**Great Lakes Exposition**—Cleveland, Ohio, June 29–Oct. 4. Lincoln G. Dickey, manager, Terminal Tower, Cleveland, Ohio.

**National Electrical and Radio Exposition**—Madison Square Garden, Sept. 18–28. Joseph Bernhard, managing director, Grand Central Palace, New York, N. Y.

**National Metals Congress & Exposition** — 1936 Annual meeting, Cleveland, Ohio, Oct. 19-23. W. H. Eisenman, secretary, 7016 Euclid Ave., Cleveland, Ohio. The following national societies will participate: American Welding Society, the Wire Association, the Institute of Metals, the American Society for Metals, the American Institute of Mining & Metallurgical Engineers, and the American Society of Mechanical Engineers.

## *Do You Know That——*

• Steel, iron and non-ferrous metals are now being cast in cement bonded molds. It is claimed that the resulting castings conform more closely to the shape of the pattern, are more dense, have a smoother surface and require less machining. (24)

• Short exposures to radiations of special rays will kill bacteria, fruit flies and cockroaches. Common salt exposed to these rays turns yellow and then black. Starch and glycerine becomes acid upon exposure and castor oil forms an insoluble compound. (25)

• Steel springs subjected to corrosion-fatigue conditions may have their endurance limits increased as much as 100 per cent by an electroplated coating of cadmium 0.0002 in. thick. (26)

*(For details, write the editor, giving the item number.)*

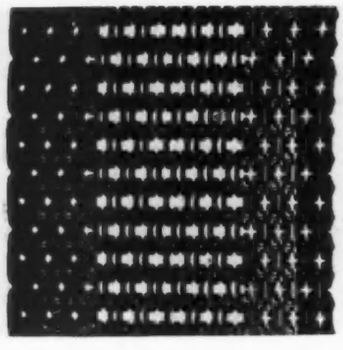
# New Materials and Parts

## *Apex Slide Rule*

Circular calculator has a C scale 25 in. long. Has regular and a number of special scales not found on other slide rules. Log-log scale is divided from 1.15 to 1,000,000. Binary scale is divided into 64ths, 32nds, and 16ths, so that fractions and mixed numbers can be multiplied and divided without changing them to decimals. Another scale gives sizes in decimals, fractions, millimeters, letter, or series number for drills to use for threads from 3 to 50 per inch, either V or U. S. Standard. Eleven engine-divided scales are on the front side and two hair-line indicators for close reading. Scales on the back give sines, cosines, tangents and cotangents for all degrees, also decimal equivalents of fractions to six figures. The calculator is 8¼ in. in diameter. Gilson Slide Rule Co., Stuart, Fla.

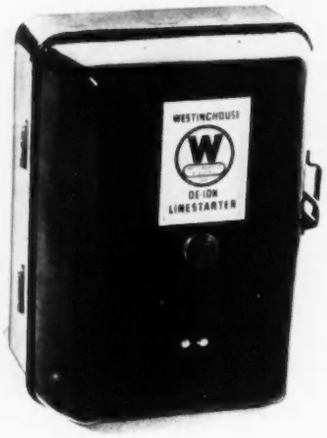
## *Flat Conveyor Chain*

Smooth open-mesh metallic-belt surface for uniform support. Particularly designed for conveying glass from molding machine to annealing oven, to allow flow of cooling blast through chain. Can be used wherever a flat metallic belt is desired. Available in ½ in. pitch, and nominal widths from 3½ to 8 in., made entirely of steel, or steel with bronze center links, or stainless steel. Driven by wide face silent chain sprocket which engages center section. Whitney Chain & Mfg. Co., Hartford, Conn.



## *De-ion Linestarters*

Suitable for across-the-line starting for single and polyphase squirrel cage motors and as primary switches for wound rotor induction motors. It is

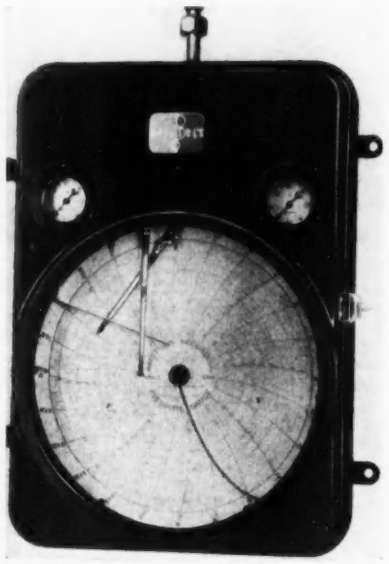



claimed the De-ion arc quencher confines, divides and extinguishes the arc almost instantaneously with unusually high rupturing ability. The arc box is a one-piece non-carbonizing porcelain section with inspection openings. All parts are accessible from the front, are self-insulated and mounted on a steel base. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

## *New Metal Finish*

For machinery and industrial equipment. It has a modified asphalt base. The following properties are claimed for the product: insoluble in oil and the lighter distillates of petroleum, is not affected by dilute caustic soda, soap and other alkaline solutions, acetic, dilute sulphuric and hydrochloric acids. Also resistant to the effects of water, sunlight, weather and fire. Known as "Aspro Metal Finish" it may be applied by spray gun or brush and may be baked or air-dried. Obtainable in black and the darker shades of gray, brown, red and green. Asphalt Products Co., Inc., Grand Central Terminal, New York, N. Y.

## *Recording-Thermometer*

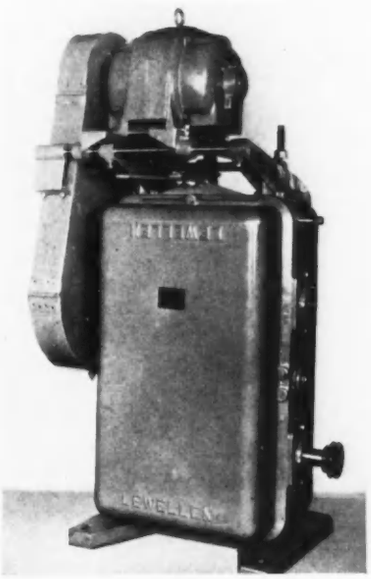


Small-bulb gas-filled for measuring temperatures between —60 and 1,000 deg. F. where tubing lengths as long as 100 ft. are required. Compensated for temperature changes in the housing and along the tubing. Consists of a rectangular case in which are mounted two measuring elements connected through a compensating mechanism. Small-bulb class 111 thermometers are offered with one or two pens in the 12-in. chart size; and with one pen only in the 8-in. chart size. The Bristol Co., Waterbury, Conn.

## *Motoraser*

Electric erasing machine for drawings. Fits in the hand, weighs six ounces, has a finger tip switch. Mech-

anism is contained in a two-piece housing of molded composition. Made in a.c. and d.c. models. Keuffel & Esser Co., Hoboken, N. J.

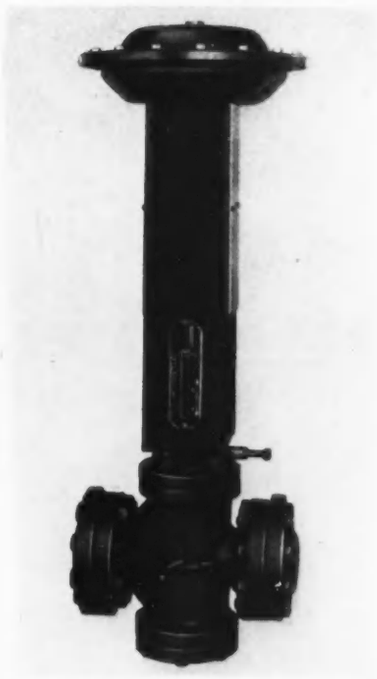


## *Variable-Speed Drive*

With speed change lever between motor and transmission. The lever can extend in any direction and can be moved either manually or by a connecting machine element so that for any position a definite speed is obtained. Applicable to both open and inclosed transmissions and in both horizontal and vertical types. Lewellen Mfg. Co., Columbus, Ind.

## *Motosteel Valve*

Formed impressions in the top plate provide free access of air pressure to the diaphragm in starting. Has a long spring of large diameter to give great initial compression and accurate response. Roller-bearing stem guide insures a free rolling action throughout the entire stroke. Spring, adjusting nut and other moving parts are fully inclosed. Side plates are removable for adjustments. Position indicator shows the valve opening. All parts are specially treated to resist corrosion. Taylor Instrument Companies, Rochester, N. Y.



## *Grasselli Bright Zinc*

New type of zinc electro-deposit in a brilliant, dense and ductile form. The ornamental finish produced is claimed to have eye-appeal and to be highly rust resistant. The improvements are the result of molybdenum. New process operates at normal room temperatures, high current efficiencies, high or low current densities, with a plating range from 15 to 100 amp. per sq.ft. Grasselli Chemical Co., 621 Euclid Ave., Cleveland, Ohio.

## *Supero Siameez Hose*

Cross-section resembles the figure 8. Twin hose are simultaneously molded with a connecting web between them. Especially designed for welding equipment. Connecting web is cut away when attaching hose to separate outlet. Special construction permits the use of different pressures in each side, without twist or writhing. Made with individual hose in different colors, or with both hose lines the same color. Electric Hose & Rubber Co., Wilmington, Del.

## *Graphic Lamacoid*

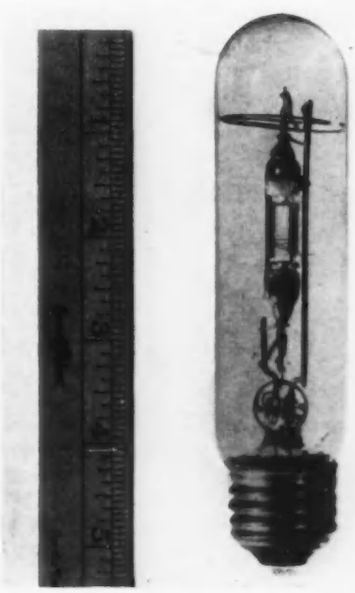
Incorporates designs, colors, printed matter, diagrams and reproductions into translucent or opaque sheets of laminated plastic material as a surface of the finished panel. Can be employed for signs, window displays, dials, charts, panels, and wiring diagrams. A variety of sizes and thicknesses are available. Mica Insulator Co., New York, N. Y.

## *Front Operated Switch*

Two styles, one a two pole, and the other a two pole solid neutral. For controlling motors heavy lighting circuits, and heating devices. Have double break contacts, can be locked in either "on" and "off" positions. The type "D" switches are rated at 30 amp., 125 volts, and are for plug fuses. Trumbull Electric Mfg. Co., Plainville, Conn.

## *New Mercury Lamp*

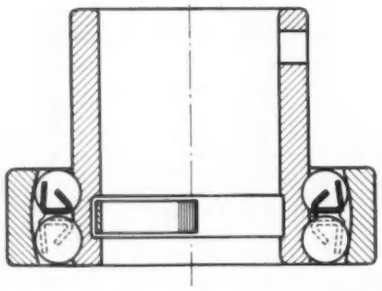
Source of light is a minute column of mercury vapor sealed in a quartz tube. Initial application will be in scientific and industrial uses where small size is desirable. Light possesses color characteristics generally similar to other mercury lamps, except for somewhat more red radiation. Rated at 85-watts and an output of 3,000 lumens, the unit is mounted within a standard T-10 bulb ($5\frac{1}{2}$ in. long and $1\frac{1}{4}$ in. diameter) with stand-



ard screw base. Operates through a ballasting type of transformer. General Electric Vapor Lamp Co., Hoboken, N. J.

## *Grip-Lock Bearing*

Positive lock type, quick application without tools. Self-aligning extended inner race bearing has eccentric groove in bore, fitted with spring steel grip-lock shoe. The bearing is slipped on shaft when shoe is in deepest part of groove. Holding the inner



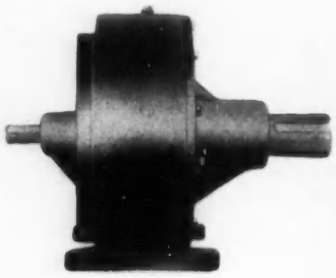
race while the shaft turns causes the knurled shoe ends to wedge in shallow part of groove, locking bearing on shaft. SKF Industries, Inc., Front St. and Erie Ave., Philadelphia, Pa.

## *Mercury Switch*

Has a metallic envelope encased in a plastic molded insulator, and operates with mercury to mercury contact. A liquid fill above the mercury is said to eliminate hydrogen gas leakage. Other desirable features claimed are minimum operating angle, improved heat radiation, long operating life, and uniform points of opening and closing the circuit. May be installed with any side up. Available in 5, 20, 50 and 200 amp. capacities. Durakool, Inc., 1010 N. Main St., Elkhart, Ind.

## *Planetary Reducer*

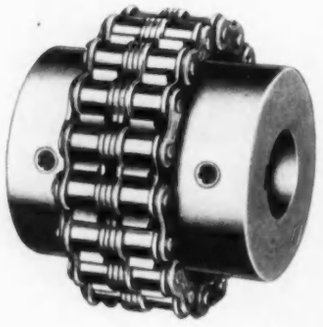
Differential type with compact straight line drive. To each planetary gear is keyed or integrally compounded a secondary gear. A freely revolving internal gear meshes with all the secondary gears. Alignment is maintained by a carrier which com-



bines the planetary gears with their journals in a detachable sub-assembly. Anti-friction bearings are used. Ratios are available from 20 to 1, and up. Winfield H. Smith, Inc., Springville, Erie County, N. Y.

## *Chain Type Flexible Coupling*

Consists of two sprockets encircled by a length of double-strand roller chain. Clearance is provided between the sprocket teeth and the chain side plate to permit misalignment of the connected shafts. This new design increases the hp. capacity of a given size coupling, also permits for a given outside diameter its use on larger shafts than the former single-strand chain coupling. Diamond Chain & Mfg. Co., Indianapolis, Ind.



## *Armco Stabilized Steel*

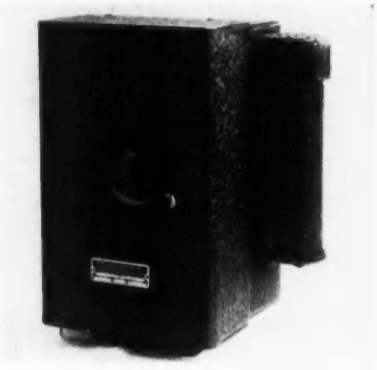
Offered as a uniform deep-drawing non-aging cold rolled steel with all stretcher strain permanently eliminated in the tempered condition. Claimed to retain indefinitely all the properties of temper-rolled steel, making pre-frabrication treatments unnecessary, regardless of the length of time the metal has been in stock. Has no sharp yield point in the temper rolled condition. Manufacturers using high finish deep drawing steel sheets will be interested in this development. American Rolling Mill Co., Middletown, Ohio.

## *Self-Aligning Pillow Block*

Operates on a true ball and socket principle. The spherical interior surface of frame is machined to receive the box with a fit that permits an oscillating movement of about three degrees in any direction. It is furnished with a Hyatt heavy-duty precision bearing in a dust and dirt proof housing. A feature claimed is its ability to withstand a greater shock load than a shaft of corresponding diameter. It is made in standard shaft and special sizes. Palmer-Bee Co., Detroit, Mich.

## *Direct Current Photo-Electric Relay*

Operating on 115 and 230 volts, for industrial control and commercial applications. Resistor unit is mounted on side. Unit illustrated is No. 5353-C with 2 in. lens aperture. Other units are available with 3 in. lens aperture, or rectangular cell aperture and no lenses, or with no apertures. Steel cabinet houses the device. G-M Laboratories, Inc., Chicago, Ill.



## *Photo-Electric Relay*

Unit for use on 110-120 volt, 50-60 cycle service, also available in other a.c. voltages and frequencies. The No. 5351-D relay illustrated uses a 3 in. lens aperture. Models are obtainable with 2 in. lenses, with rectangular

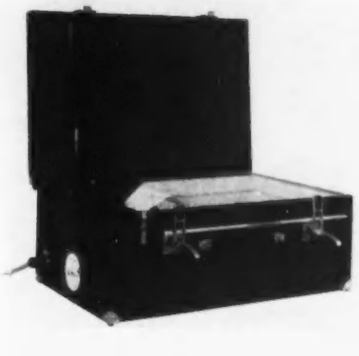
apertures and no lenses, and with no apertures for use with separate photo tube housing. The apparatus is housed in a steel cabinet. G-M Laboratories, Inc., Chicago, Ill.

## *Hipress Air Hose*

Designed for all types of air tool applications. The inner wall structure consists of four plies of duck, the outer wall is a tight braid of high tensile cords applied with tension to give flexibility and to secure adhesion of cover. Between the two walls is an insulation which serves as a secondary tube to seal off penetration of air and to cushion outside blows. The tube is made of a special rubber capable of resisting both oil and heat. Furnished in three sizes, ½, ¾ and 1in. and is made in 500 ft. lengths. B. F. Goodrich Co., Akron, Ohio.

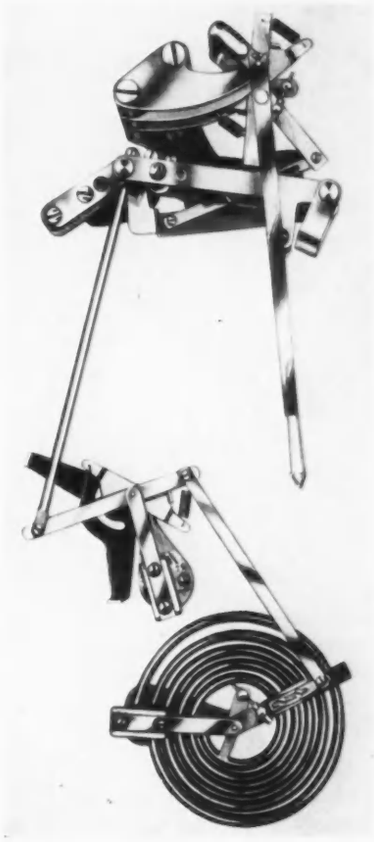
## *Electro-Copyist*

Method for reproducing originals. For photo-copying letters, documents, orders, specifications, layouts, plans, sketches and calculations. Has automatic timing clock to regulate exposure, and a finger switch to operate two tubular lighting units which supply special actinic light. White enameled parabolic reflectors direct light through a diffusion plate over the original and copy. Uses ordinary a.c. or d.c. current. Focusing is not necessary since there is no lens. The sensitive paper is put up in rolls, and after exposure is developed and fixed in a special chemical bath. Hunter Electro-Copyist, Inc., Syracuse, N. Y.



## *Pressure Compensator*

Automatically corrects errors caused by variations in static pressure in the measurement of the flow of compressible fluids. Consists of a spiral pressure tube connected to a lever, as the line pressure changes, the spiral adjusts the multiplication of linkage between the float and pen-arm. The linear amount of correction differs at different points of the chart and is calibrated to suit the flow of the meter. Foxboro Co., Foxboro, Mass.



# MANUFACTURERS' PUBLICATIONS

**Fittings for Pipe Welding** — Tube-Turns, Inc., 425 Fifth St., Louisville, Ky. Catalog and price list No. 107, 48 pages, 8½x11 in. General information covering construction features and properties of seamless steel drawn fittings for pipe welding. Contains 16 pages of dimension and weight tables covering turns, tees, flanges, heads, reducers, and shaped nipples. Concludes with an illustrated section of installations showing uses of the products.

**Forged Steel Valves** — Hancock Valve Div., Consolidated Ashcroft Hancock Co., Inc., Bridgeport, Conn. Bulletin 7,000, 20 pages, 7⅞x10¾ in. Describes new line of globe and check valves for all steam pressure ranges up to 1,500 lb. Dimensions of screwed and flanged end types, with material specifications, are given with line cut sectional views. Valves made of bar stock are also shown. Service ratings for steam and hot oil are listed.

**Instruments for Measurement and Control** — G. M. Laboratories, Inc., 1731-35 Belmont Ave., Chicago, Ill. Catalog "Instruments for Measurement and Control," 16 pages, 8½x11 in. Illustrating and describing rheostats, galvanometers, photo-electric tubes and accessories, relay, laboratory and shop appliances, used primarily in educational, research, and industrial laboratories. Capacities and characteristics are given.

**Motorized Speed Reducers**—Janette Mfg. Co., Chicago, Ill. Bulletin 22-5, 4 pages, 8½x11 in. Describes two lines of speed reducers, one a combination single reduction worm gear and planetary spur gear unit, the other a multiple spur gear type. Illustrates some standard types and shows details of construction.

**Oxwelding Brass and Bronze**—Linde Air Products Co., 30 East 42d St., New York, N. Y. Bulletin, 12 pages, 8½x11 in. Treats of the technique used in ornamental work to match and maintain appearance same as base metal, by adjusting flame and using correct welding rod. Photomicrographs of weld and base metal are included. A section describes joint design, material consumption, and tensile strength of welding on yellow brass pipe.

**Speed-Trol Motors** — Sterling Electric Motors, Inc., Los Angeles, Calif. Bulletin No. 139, 4 pages, 8½x11 in. Announcing the improved line formerly known as Sterling Vari-Speed Motors. Contains list of improvements, description of drive shown in phantom, and cuts showing standard mounting arrangements.

**Universal Joint**—The Borgeson Mfg. Co., Torrington, Conn. Folder, 8 pages, 3½x6¼ in. The bulletin stresses one of the outstanding features of this universal joint in that the blocks and angle pins are adjustable for wear. Cross sectional and exploded views show the construction of the joint. A table of various sizes of joints from ⅜ to 2⅞ in length of 1¼ to 7½ in., and for shafts ¼ to 1¾ in. is also included.

# BOOKS AND BULLETINS

## *Glances At Industrial Research*

*Edward R. Weidlein and William A. Hamor.* 246 pages, 5¼x7¾ in. Red clothboard covers. Published by the Reinhold Publishing Corp., 330 West 42d St., New York, N. Y. Price $2.75.

As the title implies, the principal aims of this book are to aid in developing technology by advancing manufacturing operations through the application of scientific methods and by discovering new materials, new processes and new uses for products. It points out that scientific management should seek collaboration of scientific research.

It is written especially for business executives and shows how progressive companies improve their present commodities and their methods of manufacture. Types of research are defined, such as research on plastics, chemical products from acetylene, and facts on Plaskon, Calgon, Coblac, and Robertson bonded metal. The creation of new industries, how new branches of manufacture have opened through research are discussed. Notes on chemical discoveries and inventions are included.

## *Standards for Rotating Electrical Machinery*

*American Standards Association,* 29 West 39th St., New York, N. Y. Paper covers, 97 pages, 7¾x10¾ in. Price $1.30.

Revisions of material formerly published by the American Institute of Electrical Engineers are combined with motor and generator standards of the National Electrical Manufacturers Association, to which new subject matter has been added by the American Standards Association Committee. It contains classifications, standard requirements as to service conditions, heating and torque values with specifications for direct current, synchronous, and induction machines, synchronous converters and fractional horsepower motors. The book begins with an important section on the definition of terms and stipulations that characterize the rating and performance of electrical machinery, with special reference to the conditions upon which acceptance tests are based. The appendix deals with lettering of dimension sheets, form for characteristic motor curves, brushes, ammonia and air compressor factors.

These are the first standards approved as a result of the work of a sectional committee on Rotating Electrical Machinery. They mark an important step in the progress of the standardization of electrical apparatus.

## *Die-Casting*

*Charles O. Herb,* 300 pages, 178 illustrations, 6x9 in. Black clothboard covers. Published by the Industrial Press, 148 Lafayette St., New York, N. Y. Price $3.

A comprehensive treatise on the art of die casting, not only in the softer metals but also in brass and cast iron. It dwells at length on alloys, die design and machines and covers the die-casting process from its beginning up to the present day practices. Photographs illustrate both thin and heavy section castings and numerous line drawings show the various types of machines in cross section.

The physical and chemical characteristics are given for die-casting materials of aluminum, magnesium, zinc, brass, bronze, manganese-bronze, aluminum-bronze, lead-antimony and tin. Die-casting dies are discussed, accompanied with a number of illustrations to show the design of single and multiple dies, exterior core work, cores in both die sections, adjustable dies for parts of different dimensions and standards of design. A chapter of 23 pages gives data on brass die casting. Another chapter on cast iron die-casting includes the most recent developments of this new process. Examples of work are shown and a picture of the apparatus and die design data are also included.

• • •

### Tin Research

Eight recent reports, published by the International Tin Research Development Council. Copies are available free from L. J. Tavener, 149 Broadway, New York, N. Y.

"The Corrosion of Tinplate," Series A, No. 30, 12 pages.

"The Constitution of the Tin-Rich Antimony—Tin Alloys," Series A, No. 31, 14 pages.

"Influence of Surface Cuprous Oxide Inclusions on the Porosity of Hot-Tinned Coatings on Copper," Series A, No. 32, 10 pages.

"The Hot-Tinning of Copper: the Attack on the Basis Metal and Its Effects," Series A, No. 33, 10 pages.

"Some Recent Investigations on the Corrosion of Tin," Series A, No. 34, 14 pages.

"Research on Thin Layers of Tin and Other Metals: II. The Corrosion of Metals by Technical Insulating Oils," Series A, No. 35, 18 pages.

"Methods of Determining Tin Plate for Examination of the Thickness and Continuity of the Alloy Layer," Series A, No. 37, 10 pages.

"A Study of the Origin of Porosity in the Tin Coating of Tin Plate," Series A, No. 38, 15 pages.

### Corrosion-Protective Value of Electrodeposited Zinc and Cadmium Coatings on Steel

*William Blum, Paul W. C. Strausser, and Abner Brenner,* National Bureau of Standards, Research Paper RP 867. 30 pages. Obtainable from Superintendent of Documents, Government Printing Office, Washington, D. C. Price 5 cents.

Report of investigation to determine relative protective values of zinc and cadmium coatings on steel. Tests covering 3½ years continuous exposure in rural or marine climates, showed thin coatings of these materials furnished better protection than did equally thin coatings of nickel or chromium. In an industrial atmosphere where sulphurous and sulphuric acids were present, both zinc and cadmium were attacked rapidly and it was found that the life was proportional to the coating thickness. Zinc-cadmium alloys containing approximately 10 per cent of cadmium were superior to either zinc or cadmium. Hot-dipped zinc coatings gave about the same protection as plated zinc coatings of the same thickness. Accelerated and other tests and results are also discussed.

### An Introduction to Springmaking

*The Wallace Barnes Co.,* Bristol, Conn. Bulletin, 52 pages, 8½x10¾ in. Paper cover.

The purpose of this publication is to promote a better understanding of springs among those who use and design them. Some of the material has been condensed from issues of the house organ "The Mainspring," to which have been added descriptions of new methods and developments.

It contains many discussions, with specific examples, of the types and applications of springs in every-day use. Much of the design information is charted. Formulas are given for obtaining correct specifications for springs of various shapes and different loading conditions. Complete tables of physical properties with photographs of the microstructure of spring materials are included. A page is devoted to the subject of corrosion prevention, another treats of the effect of temperature on spring materials.

As a handy reference book of spring information it should be of value to designing and product engineers. It is generously illustrated with line cuts and photographs.

### Graphical Symbols

*American Standards Association,* 29 W. 39th St., New York, N. Y. Bulletin Z 14.2, 16 pages, 8x10½ in. Price 45 cents.

Contains standard graphical symbols for drawings and drafting room practice covering plumbing, piping, pipe fittings and valves, heating and ventilating units, and heat-power apparatus. Also rivets, electric power apparatus and wiring arrangements. This material was assembled from publications of other organizations and revised by the Sectional Committee on Standards for Drawings and Drafting Room Practice.

# *Slider-Crank Mechanism*

J. L. SWYERS
*Atlas Imperial Diesel Engine Company*

THE accompanying tables, although primarily intended for use in calculations for gas, diesel and steam engines, air compressors and ice machines, are applicable to any slider-crank mechanism.

The distance the slider has traveled from its extreme position is expressed in the series:

$$s = r(q_0 + q_1 \cos\Theta + q_2 \cos 2\Theta + q_4 \cos 4\Theta + q_6 \cos 6\Theta + q_8 \cos 8\Theta + \ldots\ldots)$$

$$q_0 = 1 + K/4 = 3K^3/64 + 5K^5/256 + 175K^7/16{,}384 + \ldots\ldots)$$

$$q_1 = -1$$

$$q_2 = -(K/4 + K^3/16 + 15K^5/512 + 35K^7/2{,}048 + \ldots\ldots)$$

$$q_4 = (K^3/64 + 3K^5/256 + 35K^7/4{,}096 + \ldots\ldots)$$

$$q_6 = -(K^5/512 + 5K^7/2{,}048 + \ldots\ldots)$$

$$q_8 = (5K^7/16{,}384 + \ldots\ldots)$$

$\Theta$ = crank angle measured from extreme position

$K = r/l$ = ratio of length of crank, to center to center length of connecting rod

Letting $F_1 = (q_0 + q_1 \cos\Theta + q_2 \cos 2\Theta + q_4 \cos 4\Theta + q_6 \cos 6\Theta + q_8 \cos 8\Theta + \ldots\ldots)$,

the equation for the slider position becomes simply:

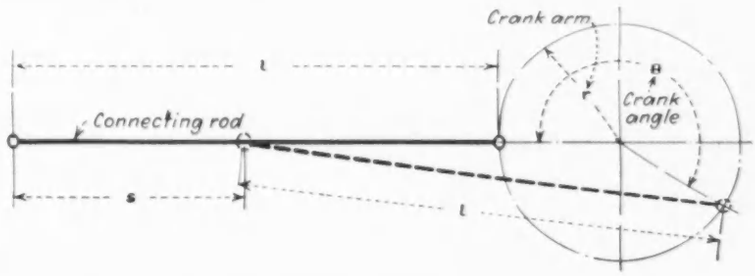
$$s = rF_1 \qquad (1)$$

$s$ = distance slider has traveled from its extreme position

$F_1$ = factor from table

$r$ = crank radius

Values of $r$ in in. gives $s$ in in. while $r$ in ft. gives $s$ in ft.




## *Table I — Factors for Determining Slider Location from Extreme Position*

| Crank Angle in Deg. | RATIO OF LENGTH OF ROD TO LENGTH OF CRANK | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 3.5 | 3.6 | 3.7 | 3.8 | 3.9 | 4.0 | 4.1 | 4.2 | 4.3 | 4.4 | 4.5 | 4.6 | 4.7 | 4.8 | 4.9 | 5.0 |
| 0–360 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 |
| 5–355 | 0.005 | 0.005 | 0.005 | 0.005 | 0.005 | 0.005 | 0.005 | 0.005 | 0.005 | 0.005 | 0.005 | 0.005 | 0.005 | 0.005 | 0.005 | 0.005 |
| 10–350 | 0.020 | 0.019 | 0.019 | 0.019 | 0.019 | 0.019 | 0.019 | 0.019 | 0.019 | 0.019 | 0.019 | 0.019 | 0.018 | 0.018 | 0.018 | 0.018 |
| 15–345 | 0.044 | 0.043 | 0.043 | 0.043 | 0.043 | 0.043 | 0.042 | 0.042 | 0.042 | 0.042 | 0.042 | 0.041 | 0.041 | 0.041 | 0.041 | 0.041 |
| 20–340 | 0.077 | 0.077 | 0.076 | 0.076 | 0.075 | 0.075 | 0.075 | 0.074 | 0.074 | 0.074 | 0.073 | 0.073 | 0.073 | 0.073 | 0.072 | 0.072 |
| 25–335 | 0.119 | 0.119 | 0.118 | 0.117 | 0.117 | 0.116 | 0.116 | 0.115 | 0.115 | 0.114 | 0.114 | 0.113 | 0.113 | 0.112 | 0.112 | 0.112 |
| 30–330 | 0.170 | 0.169 | 0.168 | 0.167 | 0.166 | 0.165 | 0.165 | 0.164 | 0.163 | 0.163 | 0.162 | 0.161 | 0.161 | 0.160 | 0.160 | 0.159 |
| 35–325 | 0.228 | 0.227 | 0.226 | 0.224 | 0.223 | 0.222 | 0.221 | 0.220 | 0.219 | 0.218 | 0.218 | 0.217 | 0.216 | 0.215 | 0.215 | 0.214 |
| 40–320 | 0.294 | 0.292 | 0.290 | 0.289 | 0.287 | 0.286 | 0.285 | 0.283 | 0.282 | 0.281 | 0.280 | 0.279 | 0.278 | 0.277 | 0.276 | 0.275 |
| 45–315 | 0.365 | 0.363 | 0.361 | 0.359 | 0.358 | 0.356 | 0.354 | 0.353 | 0.351 | 0.350 | 0.349 | 0.348 | 0.346 | 0.345 | 0.344 | 0.343 |
| 50–310 | 0.442 | 0.440 | 0.437 | 0.435 | 0.433 | 0.431 | 0.429 | 0.428 | 0.426 | 0.424 | 0.423 | 0.421 | 0.420 | 0.419 | 0.418 | 0.416 |
| 55–305 | 0.524 | 0.521 | 0.518 | 0.516 | 0.513 | 0.511 | 0.509 | 0.507 | 0.505 | 0.503 | 0.502 | 0.500 | 0.498 | 0.497 | 0.495 | 0.494 |
| 60–300 | 0.609 | 0.606 | 0.603 | 0.600 | 0.597 | 0.595 | 0.593 | 0.590 | 0.588 | 0.586 | 0.584 | 0.582 | 0.581 | 0.579 | 0.577 | 0.576 |
| 65–295 | 0.697 | 0.693 | 0.690 | 0.687 | 0.684 | 0.681 | 0.679 | 0.676 | 0.674 | 0.672 | 0.670 | 0.668 | 0.666 | 0.664 | 0.662 | 0.660 |
| 70–290 | 0.787 | 0.783 | 0.779 | 0.776 | 0.773 | 0.770 | 0.767 | 0.764 | 0.762 | 0.760 | 0.757 | 0.755 | 0.753 | 0.751 | 0.749 | 0.747 |
| 75–285 | 0.877 | 0.873 | 0.870 | 0.866 | 0.863 | 0.860 | 0.857 | 0.854 | 0.851 | 0.849 | 0.846 | 0.844 | 0.842 | 0.839 | 0.837 | 0.835 |
| 80–280 | 0.968 | 0.964 | 0.960 | 0.956 | 0.953 | 0.950 | 0.946 | 0.943 | 0.941 | 0.938 | 0.935 | 0.933 | 0.931 | 0.929 | 0.926 | 0.924 |
| 85–275 | 1.058 | 1.053 | 1.050 | 1.046 | 1.042 | 1.039 | 1.036 | 1.033 | 1.030 | 1.027 | 1.025 | 1.022 | 1.020 | 1.017 | 1.015 | 1.013 |
| 90–270 | 1.146 | 1.142 | 1.138 | 1.134 | 1.130 | 1.127 | 1.124 | 1.121 | 1.118 | 1.115 | 1.113 | 1.110 | 1.108 | 1.105 | 1.103 | 1.101 |
| 95–265 | 1.232 | 1.228 | 1.224 | 1.220 | 1.217 | 1.213 | 1.210 | 1.207 | 1.204 | 1.201 | 1.199 | 1.196 | 1.194 | 1.192 | 1.190 | 1.187 |
| 100–260 | 1.315 | 1.311 | 1.307 | 1.304 | 1.300 | 1.297 | 1.294 | 1.291 | 1.288 | 1.285 | 1.283 | 1.280 | 1.278 | 1.276 | 1.274 | 1.272 |
| 105–255 | 1.395 | 1.391 | 1.387 | 1.384 | 1.380 | 1.377 | 1.374 | 1.371 | 1.369 | 1.366 | 1.364 | 1.361 | 1.359 | 1.357 | 1.355 | 1.353 |
| 110–250 | 1.471 | 1.467 | 1.463 | 1.460 | 1.457 | 1.454 | 1.451 | 1.449 | 1.446 | 1.444 | 1.441 | 1.439 | 1.437 | 1.435 | 1.433 | 1.431 |
| 115–245 | 1.542 | 1.539 | 1.535 | 1.532 | 1.529 | 1.527 | 1.524 | 1.522 | 1.519 | 1.517 | 1.515 | 1.513 | 1.511 | 1.509 | 1.507 | 1.505 |
| 120–240 | 1.609 | 1.606 | 1.603 | 1.600 | 1.597 | 1.595 | 1.593 | 1.590 | 1.588 | 1.586 | 1.584 | 1.582 | 1.581 | 1.579 | 1.577 | 1.576 |
| 125–235 | 1.671 | 1.668 | 1.665 | 1.663 | 1.661 | 1.658 | 1.656 | 1.654 | 1.652 | 1.651 | 1.649 | 1.647 | 1.646 | 1.644 | 1.643 | 1.641 |
| 130–230 | 1.728 | 1.725 | 1.723 | 1.721 | 1.719 | 1.717 | 1.715 | 1.713 | 1.712 | 1.710 | 1.709 | 1.707 | 1.706 | 1.704 | 1.703 | 1.702 |
| 135–225 | 1.779 | 1.777 | 1.775 | 1.774 | 1.772 | 1.770 | 1.769 | 1.767 | 1.766 | 1.764 | 1.763 | 1.762 | 1.761 | 1.760 | 1.758 | 1.757 |
| 140–220 | 1.826 | 1.824 | 1.822 | 1.821 | 1.819 | 1.818 | 1.817 | 1.816 | 1.814 | 1.813 | 1.812 | 1.811 | 1.810 | 1.809 | 1.808 | 1.808 |
| 145–215 | 1.867 | 1.865 | 1.864 | 1.863 | 1.862 | 1.861 | 1.860 | 1.859 | 1.858 | 1.857 | 1.856 | 1.855 | 1.854 | 1.854 | 1.853 | 1.852 |
| 150–210 | 1.902 | 1.901 | 1.900 | 1.899 | 1.898 | 1.897 | 1.897 | 1.896 | 1.895 | 1.895 | 1.894 | 1.893 | 1.893 | 1.892 | 1.892 | 1.891 |
| 155–205 | 1.932 | 1.931 | 1.931 | 1.930 | 1.929 | 1.929 | 1.928 | 1.928 | 1.927 | 1.927 | 1.926 | 1.926 | 1.925 | 1.925 | 1.925 | 1.924 |
| 160–200 | 1.956 | 1.956 | 1.956 | 1.955 | 1.955 | 1.954 | 1.954 | 1.954 | 1.953 | 1.953 | 1.953 | 1.952 | 1.952 | 1.952 | 1.952 | 1.951 |
| 165–195 | 1.976 | 1.975 | 1.975 | 1.975 | 1.975 | 1.974 | 1.974 | 1.974 | 1.974 | 1.974 | 1.973 | 1.973 | 1.973 | 1.973 | 1.973 | 1.973 |
| 170–190 | 1.989 | 1.989 | 1.989 | 1.989 | 1.989 | 1.989 | 1.989 | 1.988 | 1.988 | 1.988 | 1.988 | 1.988 | 1.988 | 1.988 | 1.988 | 1.988 |
| 175–185 | 1.997 | 1.997 | 1.997 | 1.997 | 1.997 | 1.997 | 1.997 | 1.997 | 1.997 | 1.997 | 1.997 | 1.997 | 1.997 | 1.997 | 1.997 | 1.997 |
| 180–180 | 2.000 | 2.000 | 2.000 | 2.000 | 2.000 | 2.000 | 2.000 | 2.000 | 2.000 | 2.000 | 2.000 | 2.000 | 2.000 | 2.000 | 2.000 | 2.000 |

# *Slider-Crank Mechanism*

J. L. SWYERS

*Atlas Imperial Diesel Engine Company*

TABLE II gives the factors for the velocity of the slider at different crank positions. Since the slider velocity is the derivative of the slider position with relation to time the following equations are evolved:

$$v = -r(d\theta/dt)\,(q_1 \sin\theta + 2q_2 \sin 2\theta + 4q_4 \sin 4\theta + 6q_6 \sin 6\theta + 8q_8 \sin 8\theta + \ldots\ldots)$$

$$v = r(d\theta/dt)F_2 \qquad (2)$$

$$F_2 = -(q_1 \sin\theta + 2q_2 \sin 2\theta + 4q_4 \sin 4\theta + 6q_6 \sin 6\theta + 8q_8 \sin 8\theta + \ldots\ldots)$$

Assuming $d\theta/dt = \omega$ which is the average angular velocity of the crank, the above equation may be written as:

$$v = r\omega F_2 \qquad (3)$$

In the following equation for slider velocity, $v = F_2\, r\, (2\pi N/60)$, $r$ in in. gives $v$ in in. per min. while $r$ in ft. gives $v$ in ft. per min.

In the equation for the crank moment, $M = F_2\, Pr$, when $r$ is given in in. $M$ will be in in. lb. and $r$ in ft. gives $M$ in ft. lb.

From 0 to 180 deg. all factors in the table are positive and those from 180 to 360 deg. are negative.

$v$ = velocity of slider or piston
$F_2$ = factor from table
$r$ = crank radius
$N$ = r.p.m. of crank
$M$ = crank moment
$P$ = force in lb. acting on piston

An article and four other tables of simplified calculations for reciprocating engines will appear later.

***Table II — Factors for Determining Slider or Piston Velocity and Crank Moment***

| Crank Angle in Deg. | RATIO OF LENGTH OF ROD TO LENGTH OF CRANK | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 3.5 | 3.6 | 3.7 | 3.8 | 3.9 | 4.0 | 4.1 | 4.2 | 4.3 | 4.4 | 4.5 | 4.6 | 4.7 | 4.8 | 4.9 | 5.0 |
| 0–360 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 |
| 5–355 | 0.112 | 0.111 | 0.111 | 0.110 | 0.109 | 0.109 | 0.108 | 0.108 | 0.107 | 0.107 | 0.106 | 0.106 | 0.106 | 0.105 | 0.105 | 0.105 |
| 10–350 | 0.223 | 0.221 | 0.220 | 0.219 | 0.218 | 0.216 | 0.215 | 0.214 | 0.213 | 0.213 | 0.212 | 0.211 | 0.210 | 0.209 | 0.209 | 0.208 |
| 15–345 | 0.330 | 0.328 | 0.327 | 0.325 | 0.323 | 0.321 | 0.320 | 0.319 | 0.317 | 0.316 | 0.315 | 0.313 | 0.312 | 0.311 | 0.310 | 0.309 |
| 20–340 | 0.434 | 0.432 | 0.429 | 0.427 | 0.425 | 0.423 | 0.421 | 0.419 | 0.417 | 0.415 | 0.414 | 0.412 | 0.411 | 0.409 | 0.408 | 0.406 |
| 25–335 | 0.533 | 0.530 | 0.527 | 0.524 | 0.521 | 0.519 | 0.517 | 0.514 | 0.512 | 0.510 | 0.508 | 0.506 | 0.504 | 0.503 | 0.501 | 0.500 |
| 30–330 | 0.625 | 0.621 | 0.618 | 0.615 | 0.612 | 0.609 | 0.606 | 0.604 | 0.601 | 0.599 | 0.597 | 0.595 | 0.593 | 0.591 | 0.589 | 0.587 |
| 35–325 | 0.710 | 0.706 | 0.702 | 0.699 | 0.695 | 0.692 | 0.689 | 0.687 | 0.684 | 0.681 | 0.679 | 0.677 | 0.674 | 0.672 | 0.670 | 0.668 |
| 40–320 | 0.786 | 0.782 | 0.778 | 0.774 | 0.771 | 0.768 | 0.764 | 0.761 | 0.759 | 0.756 | 0.753 | 0.751 | 0.749 | 0.746 | 0.744 | 0.742 |
| 45–315 | 0.853 | 0.849 | 0.845 | 0.841 | 0.838 | 0.834 | 0.831 | 0.828 | 0.825 | 0.822 | 0.820 | 0.817 | 0.815 | 0.812 | 0.810 | 0.808 |
| 50–310 | 0.910 | 0.906 | 0.902 | 0.898 | 0.895 | 0.892 | 0.888 | 0.885 | 0.882 | 0.880 | 0.877 | 0.875 | 0.872 | 0.870 | 0.868 | 0.866 |
| 55–305 | 0.957 | 0.953 | 0.949 | 0.946 | 0.942 | 0.939 | 0.936 | 0.933 | 0.931 | 0.928 | 0.925 | 0.923 | 0.921 | 0.919 | 0.916 | 0.914 |
| 60–300 | 0.994 | 0.990 | 0.986 | 0.983 | 0.980 | 0.977 | 0.974 | 0.971 | 0.969 | 0.966 | 0.964 | 0.962 | 0.960 | 0.958 | 0.956 | 0.954 |
| 65–295 | 1.020 | 1.016 | 1.013 | 1.010 | 1.007 | 1.005 | 1.002 | 1.000 | 0.997 | 0.995 | 0.993 | 0.991 | 0.989 | 0.988 | 0.986 | 0.984 |
| 70–290 | 1.035 | 1.032 | 1.030 | 1.027 | 1.025 | 1.022 | 1.020 | 1.018 | 1.016 | 1.015 | 1.013 | 1.011 | 1.010 | 1.008 | 1.007 | 1.005 |
| 75–285 | 1.040 | 1.038 | 1.036 | 1.034 | 1.032 | 1.030 | 1.029 | 1.027 | 1.026 | 1.024 | 1.023 | 1.022 | 1.020 | 1.019 | 1.018 | 1.017 |
| 80–280 | 1.036 | 1.034 | 1.033 | 1.031 | 1.030 | 1.029 | 1.028 | 1.027 | 1.026 | 1.025 | 1.024 | 1.023 | 1.022 | 1.021 | 1.020 | 1.020 |
| 85–275 | 1.022 | 1.021 | 1.021 | 1.020 | 1.019 | 1.019 | 1.018 | 1.018 | 1.017 | 1.016 | 1.016 | 1.016 | 1.015 | 1.015 | 1.014 | 1.014 |
| 90–270 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 |
| 95–265 | 0.970 | 0.971 | 0.972 | 0.973 | 0.973 | 0.974 | 0.974 | 0.975 | 0.975 | 0.976 | 0.976 | 0.977 | 0.977 | 0.978 | 0.978 | 0.979 |
| 100–260 | 0.934 | 0.935 | 0.937 | 0.938 | 0.940 | 0.941 | 0.942 | 0.943 | 0.944 | 0.945 | 0.946 | 0.947 | 0.948 | 0.948 | 0.949 | 0.950 |
| 105–255 | 0.892 | 0.894 | 0.896 | 0.898 | 0.900 | 0.902 | 0.903 | 0.905 | 0.906 | 0.908 | 0.909 | 0.910 | 0.912 | 0.913 | 0.914 | 0.915 |
| 110–250 | 0.844 | 0.847 | 0.850 | 0.852 | 0.855 | 0.857 | 0.859 | 0.861 | 0.863 | 0.865 | 0.867 | 0.868 | 0.870 | 0.871 | 0.873 | 0.874 |
| 115–245 | 0.793 | 0.796 | 0.800 | 0.803 | 0.805 | 0.808 | 0.811 | 0.813 | 0.815 | 0.817 | 0.819 | 0.821 | 0.823 | 0.825 | 0.827 | 0.828 |
| 120–240 | 0.738 | 0.742 | 0.746 | 0.749 | 0.752 | 0.755 | 0.758 | 0.761 | 0.763 | 0.766 | 0.768 | 0.770 | 0.772 | 0.774 | 0.776 | 0.778 |
| 125–235 | 0.681 | 0.685 | 0.689 | 0.693 | 0.696 | 0.699 | 0.702 | 0.705 | 0.708 | 0.711 | 0.713 | 0.715 | 0.718 | 0.720 | 0.722 | 0.724 |
| 130–230 | 0.622 | 0.626 | 0.630 | 0.634 | 0.637 | 0.641 | 0.644 | 0.647 | 0.650 | 0.652 | 0.655 | 0.658 | 0.660 | 0.662 | 0.664 | 0.666 |
| 135–225 | 0.561 | 0.566 | 0.570 | 0.573 | 0.577 | 0.580 | 0.583 | 0.586 | 0.589 | 0.592 | 0.595 | 0.597 | 0.600 | 0.602 | 0.604 | 0.606 |
| 140–220 | 0.500 | 0.504 | 0.508 | 0.511 | 0.515 | 0.518 | 0.521 | 0.524 | 0.527 | 0.530 | 0.532 | 0.535 | 0.537 | 0.539 | 0.541 | 0.544 |
| 145–215 | 0.438 | 0.441 | 0.445 | 0.449 | 0.452 | 0.455 | 0.458 | 0.461 | 0.463 | 0.466 | 0.468 | 0.471 | 0.473 | 0.475 | 0.477 | 0.479 |
| 150–210 | 0.375 | 0.379 | 0.382 | 0.385 | 0.388 | 0.391 | 0.394 | 0.396 | 0.399 | 0.401 | 0.403 | 0.405 | 0.407 | 0.409 | 0.411 | 0.413 |
| 155–205 | 0.312 | 0.316 | 0.318 | 0.321 | 0.324 | 0.326 | 0.329 | 0.331 | 0.333 | 0.335 | 0.337 | 0.339 | 0.341 | 0.343 | 0.344 | 0.346 |
| 160–200 | 0.250 | 0.252 | 0.255 | 0.257 | 0.259 | 0.261 | 0.263 | 0.265 | 0.267 | 0.269 | 0.270 | 0.272 | 0.274 | 0.275 | 0.276 | 0.278 |
| 165–195 | 0.187 | 0.189 | 0.191 | 0.193 | 0.195 | 0.196 | 0.198 | 0.199 | 0.201 | 0.202 | 0.203 | 0.204 | 0.206 | 0.207 | 0.208 | 0.209 |
| 170–190 | 0.125 | 0.126 | 0.127 | 0.129 | 0.130 | 0.131 | 0.132 | 0.133 | 0.134 | 0.135 | 0.136 | 0.136 | 0.137 | 0.138 | 0.139 | 0.139 |
| 175–185 | 0.062 | 0.063 | 0.064 | 0.064 | 0.065 | 0.065 | 0.066 | 0.067 | 0.067 | 0.067 | 0.068 | 0.068 | 0.069 | 0.069 | 0.069 | 0.070 |
| 180–180 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 | 0.000 |